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Recent Developments in Iron and Steel Technology*

THE first paper on trends in steel technology ("Recent Developments and Trends in Iron and Steel Technology"), was issued in July, 1952, and took the form of contributions by experts, who described some of the more interesting developments, the various individual contributions having little or no connection with each other. The present paper describes the main developments which have taken place and the direction in which new developments may be expected, together with some evaluation of their importance to the industry. The paper is based largely on articles appearing in the world technical press, as well as on specific contributions received from various experts, and has been divided into three parts, as follows :—

(1) PIG-IRON PRODUCTION AND BLAST-FURNACE PRACTICE

Some interesting developments in three completely different lines have been taking place in the field of blast-furnace practice. These developments relate to :—

(*a*) *Improved Efficiency in Conventional Blast Furnaces:* The classical problem of improving the efficiency of conventional blast furnaces is to reduce the amount of coke consumed per ton of pig iron produced and to increase output. There is also no doubt that a large furnace, with a hearth diameter of 25 feet and more, has better efficiency than a smaller unit. The effects of work in connection with these two factors are shown in the steady improvement of overall coke consumption figures during recent years, and no doubt further improvements will result from improved ore preparation, as well as the continuation of the trend to replace small furnaces by large ones.

However, the new development of "high top pressure" has recently produced interesting results. In a conventional blast furnace, the gases leave the top of the furnace at negligible pressure (less than 1 psi). The speed through the stack of descending raw materials is very high, and it is difficult to secure uniform distribution between the rising gas stream and the falling particles of the charge. Ore preparation has done much to improve these conditions, but when attempts are made in practice to increase the production of a furnace by blowing more air, thus consuming coke more quickly, the velocity of the gas is increased and the uniformity of distribution between the gas and the particles becomes progressively worse. At the same time, the amount of dust blown out of the furnace by the gas stream is increased. Another difficulty resulting from an increased quantity of air is that the rising gas stream follows the path of least resistance, thus creating in the furnace a number of chimneys in which the speed of the gas may be much greater than that of the gas passing through the charge itself.

The operation of a furnace under high top pressure requires the gases leaving the furnace to pass through a special valve, so that the top pressure is built up to about 10 psi, instead of 1 psi, as in conventional furnaces. As a result, the furnace has a far greater degree flexibility, and it can be operated in three ways, i.e.,

(i) The same rate of blowing can be maintained, thus lowering gas velocity, dust production, and consumption of coke per ton.

(ii) It can be operated at a higher rate of blowing, but, owing to the increased gas density, the gas speeds will be no higher than under normal conditions, in which case production will increase with similar flue dust losses and coke consumption as obtained under normal pressure conditions.

(iii) A compromise can be made between (i) and (ii).

(*b*) *Low-Shaft Blast Furnaces:* Much has been written and there has been considerable discussion on the development of low-shaft blast furnaces during the past few years. Two large-scale experimental furnaces have recently come into operation; one of these is a specially built furnace at Liège, with an oval hearth and an overall height of 9 metres, and the other is at Oberhausen, in Western Germany. The latter furnace is, in effect, a rebuilt standard blast furnace, with an overall height of 8·5 metres and also employing an oval hearth.

It would appear too early to state definitely that low-shaft blast furnaces have reached the stage of development when they can be regarded as a serious rival to modern large-size furnaces, with their high thermal efficiency. Oxygen-enriched blast has been used in the experimental furnaces, and low-cost oxygen is essential if this is to become standard practice.

The low-shaft blast furnace would appear to have some advantages over the standard furnace, but some disadvantages are also apparent. Before the economies of this type of furnace in large-scale production can be gauged, more information than is at present available on actual operating cost will be necessary, although preliminary results obtained with the rebuilt furnace at Oberhausen appear to show that the furnace lends itself particularly well to the production of ferro-alloys —especially ferro-manganese—where considerable reductions in coke consumption have been noted.

(*c*) *Small-Capacity Blast Furnaces:* Small-capacity blast furnaces are of particular interest to less developed countries, especially to those considering the development of a steel industry using local raw material resources which may not be of the quality necessary for large-scale operations. The development has recently taken place of a small-capacity blast furnace with a pig-iron production of 10,000 to 15,000 tons a year, using ores with a Fe-content of less than 30 per cent and using solid fuel with a heating value of under 5000 calories and with up to 30 per cent ash. A pilot plant has been in operation near Lausanne, in Switzerland, using these basically unsuitable raw materials.

In essence, the furnace consists of a small-diameter blast furnace, with a special recuperator operating at gas temperatures up to 1600° C, enabling the blast to enter the tuyères at 600° C. The interesting point of this arrangement is that the hot gases from the reduction zone, instead of being drawn upwards through the charge, as in a conventional blast furnace, are drawn downwards into the bottom of the recuperator. This enables the high temperatures to be reached and,

* Based on the publication "Some Important Developments during 1953 in Iron and Steel Technology," issued January 7, 1954, by the United Nations Economic and Social Council, and prepared by the Industry Division, Economic Commission for Europe.

furthermore, has the advantage of preventing " hanging " of the charge in the shaft.

The value of this type of unconventional reduction-furnace, as with other, better-known types, such as the rotating furnace, can be judged only after trials have been made on a pilot plant with the ores and fuel available in the country in which it is proposed to instal such equipment. This type of low-shaft furnace, owing to its simplicity and low cost, would appear to offer good chances of success with a very wide range of raw materials.

(2) STEEL MAKING

(A) The Use of Oxygen in Converters

Undoubtedly the most interesting and extensive recent development in steel making has been improvement in quality, by the use of oxygen in basic Thomas converters. Two developments have been going on at the same time, as follows :—

(*a*) *Oxygen/Steam Blowing:* Basically, the oxygen/steam blowing method can be used with any converter ; the mixture must, however, have a minimum temperature to prevent condensation of the steam. If condensation takes place, the blast becomes uncontrollable and also causes hydration in the dolomite bottom.

In oxygen/steam converter operation, the manganese in the iron is oxidised rather more rapidly than with air-blowing, and the carbon removal begins earlier. There is a slight dephosphorisation during the oxidation of silicon and manganese. Rapid decarburisation starts after two or three minutes, the rate of carbon removal amounting to 1·5 per cent per minute. After five or six minutes, the decarburisation is complete and the afterblow begins. Very little manganese reversion takes place during the change-over. During the process of conversion, there is a steady removal of nitrogen from the metal, which lasts right through the afterblow, though at a slightly reduced rate. Over 200 blows in both large and small vessels have been studied. The oxygen/steam ratio varied from 1·6 : 1 to 1 : 1·2 and the nitrogen in the metal was lowered from the region of 0·008 to 0·003 per cent.

It seems probable that the use of oxygen/steam blowing in basic Thomas converters, to improve quality, will be further developed and tried out on a more extensive scale than at present, to determine whether the method does, in fact, produce the results claimed and will prove economic.

(*b*) *The Austrian " L-D " Process:* This process consists, in the main, of blowing pure oxygen at high pressure on to the top surface of the metal in a converter. The development of this process is rather remarkable in that the time taken from the start of the initial experiments to full-scale production in a 15-ton vessel was only three and a half years.

The " L-D " process was adopted in Austria owing to peculiar conditions, in that the Austrian ores with low phosphorus lend themselves very well to melting in open-hearth furnaces, but, as the Austrian industry as a whole does not produce sufficient scrap, it has always been necessary to import steel-melting scrap to maintain open-hearth production. The use of the " L-D " process, therefore, seems to solve the Austrian problem by making it possible to convert indigenous low-phosphorus iron in a basic vessel ; the method enables the silicon and phosphorus to be removed.

Although the " L-D " process in Austria is confined to the conversion of iron which would normally be processed in the open-hearth furnace, there appears to be no reason why iron containing 1 to 2 per cent of phosphorus, instead of 0·2 per cent, might not be converted by this method. One major advantage is the fact that the cost and inconvenience of renewing the converter bottom after every 50 blows, as is the case with normal basic Thomas converters, can be avoided.

In the case of high-phosphorus ores, trials have been made at the Huckingen Steel Works, Duisburg, Germany, with the " L-D " method, in a small 3-ton converter, and have shown that high-phosphorus iron can be adequately dephosphorised to produce steel which is in every way as good as the best basic Thomas quality.

It seems clear that this process, with proper adjustment for varying grades of iron ore, can be successfully employed on a very wide range of types of iron and can produce a high quality of steel, with low nitrogen content and at favourable cost.

There are still several factors which require further development, particularly the problem of fume extraction. During the blow, heavy brown fumes are produced, and these would give rise to serious complaints from people living in the vicinity ; it seems, therefore, that this fume-extraction problem must be solved before the process can be widely introduced, particularly in densely populated areas.

(B) Open-Hearth Steel

The most interesting development in open-hearth practice is refinement in instrumentation. Much work has recently been done to reduce the dependence of open-hearth steel making on the skill of the melter.

The basic conception of the open-hearth furnace with a continuous high-temperature flue system, which enables some 50% of the heat leaving the furnace to be returned to it by constant reversals of the hot gases, makes the question of good instrumentation a difficult one to solve.

Automatic temperature control, although fairly widely used in the United States, has only comparatively recently been used extensively in Europe. The advantages claimed for instrumentation and automatic control are that they provide quantitative information on roof temperature and furnace pressure ; consequently, they enable the melter to run his furnace closer to the danger point and yet do less damage than might be expected with a lower operating level but periodic overshooting of the maximum safe temperature. The proof that this procedure pays has been provided by numerous melting shops, in which good instrumentation has resulted in higher output and has particularly improved the life of the furnace roof.

(C) Induction Stirrers

The trend towards larger arc-furnace units and their more extensive use has brought to the forefront the problem of increased efficiency. This can best be achieved by making the movement of the metal in the bath more active, by introducing a stirring effect. The two most important reasons for the desirability of this stirring action are that electric furnaces suffer from the disadvantage of having hot spots in the neighbourhood of the carbon arcs, and that the movement of the bath is naturally sluggish until an active boil is achieved.

A Swedish company has developed a stirring mechanism which functions by means of a travelling magnetic field. The application of this magnetic field to the bottom of the furnace produces induction currents, which cause the metal in the bath to flow upwards and sideways in large eddies. The operation of the stirrer is very simple and is controlled from a panel equipped with pu h-buttons for starting, stopping, and reversing. Temperature-control equipment and instruments are also provided, to check the furnace-bottom temperature.

The installation of induction stirrers in existing furnaces is probably economic only for large units, say, of 20 tons and upwards, because it means replacing the furnace-bottom plate in stainless steel instead of mild steel. It is, however, claimed that for new furnaces of about 10-ton capacity and upwards, the stirrer will pay for itself in increased output, in about 12 to 18 months. In some cases it would appear to be well worth-while installing stirrers in existing larger capacity

furnaces, particularly if such installations can coincide with the major furnace overhaul.

(3) STEEL ROLLING AND FINISHING PROCESSES

(A) Continuous Casting of Steel

The problems associated with continuous casting of steel have been a matter of research for many years. The practice of continuou · casting has been used on a commercial scale for some time in the non-ferrous metal industries, particularly aluminium, and developments have now reached the point where steel can be continuously cast satisfactorily for the production of billets or slabs, thus eliminating the need for high capital cost cogging equipment for ingot reduction.

In one of the processes now developed, machines are in operation which will produce billets of 100 × 100 mm or slabs of 120 × 500 mm ; casting speeds of up to 2·5 metres per minute are possible. No reliable figures are as yet available on production costs, but it appears that further development will be necessary before economical production is possible.

There is no question of continuous casting completely replacing blooming, cogging, and slabbing mills in the steel industry, but it may well be increasingly used for the production of higher qualities of steel, and also in works where production is comparatively modest. The method may also lend itself particularly well to the production of tube rounds.

(B) Hot-Rolling Planetary Mill

There has been an interesting development in the technique of hot-rolling steel strip. This is the so-called planetary mill, which incorporates a basically new principle to reduce, in one pass, a thick steel slab into comparatively thin, hot-rolled strip.

In all conventional types of hot-rolling, the reduction of thickness per pass is generally in the region of 20 to 30 per cent ; a reduction of up to 90 per cent in one pass can be achieved in the planetary mill.

It is of particular interest to note that instead of a drop in the temperature of the steel to be rolled, as in the case of a conventional mill, in the planetary mill the reverse is the case, the reason being that the plastic deformation of the slab in the reducing zone is such that the temperature actually rises slightly.

The planetary mill is undoubtedly an interesting attempt to solve the problem of producing hot-rolled strip from slabs with equipment of low capital cost, compared with the continuous or semi-continuous hot-strip mill. As with all unorthodox designs, certain teething troubles are to be expected but, given the high standard of maintenance which a mill of this type would require, there seems to be no reason why it should not be capable of producing hot-rolled strip, on a moderate tonnage production basis, at an economic cost.

(C) Differential Thickness Coating of Tinplate

The stage from normal electrolytic tinning, whereby the coating of tin on both sides of the steel is of the same thickness, to the stage of depositing a heavier coat on one side than on the other, is a comparatively small one. The advantages, from an economic point of view, to the tinplate user are obvious because, for many applications, a lower degree of protection is required on the outside of the tin container than on the inside.

The basic principle of electrolytic tinning is well known and, by adopting a process in which each side of the strip is plated independently of the other, it is comparatively simple, by the addition of several rheostats, to control the thickness of coating to be applied to either side. Many combinations of present commercial coating weights are possible ; these differential coatings can effect real savings in the amount of tin applied, and thus reduce the overall cost of the container.

(D) The Steel Extrusion Process

The use of the extrusion process for non-ferrous metals, particularly aluminium, has been widely developed for the production of both simple and complicated shapes as well as of tubes of all types.

The extrusion of non-ferrous metals presents few difficulties, but the extrusion of steel, on the same basic principle as that used for aluminium, has come up against the problem of die life, although tubes have been successfully extruded in the United Kingdom for many years. The possibility of producing steel bars and sections by hot extrusion has been the subject of research for over twenty years, and it has long been conclusively proved that steel is extrudable in the same way as non-ferrous metals, provided suitable lubricants can be found to prolong the life of the dies.

Development work has been most actively carried out in France, and the process is now established under the name of " Ugine-Séjournet." The results of these French developments have shown that glass used in the plastic state was not only a good lubricant but that it simultaneously provided effective insulation between the tools and the steel billet and sections being extruded. The use of glass as a lubricant and insulator brought with it many closely interrelated variables, for instance, the speed of extrusion, the temperature of the metal, the quantity of glass used, and its viscosity. However, continued experiment has finally solved these various problems and it is now possible to convert cast steel ingots, both in carbon and alloy qualities, directly into bars, sections, and tubes on a production basis.

(E) Oxygen Cutting to replace Slab and Bloom Shears

With the growing use of tonnage oxygen in steel works, the possibility of using oxygen cutting flames, to replace hot slab and bloom shears, has become of considerable interest.

The great advantage claimed for oxygen cutting is that the machine is virtually one hundred per cent available and that maintenance is negligible. The main working parts of the machine—the torch and tip—are not subject to mechanical load. All parts exposed to radiant heat are water-cooled and are fully protected by insulation. The torch tip is positioned 1½ to 2 in. above the hot steel to be cut and is therefore not easily damaged or clogged by flying scale. The tips themselves can be changed in a few minutes and the entire torch can be changed within ten minutes, if damaged.

As an example of cutting speeds possible, 8½-inch rounds have been cut consistently in seventeen seconds and, for the cutting of slabs 4 to 6 inches thick, a single torch will cut at speeds of 30 inches per minute. The use of two torches on the same cut doubles the cutting speed.

An oxygen cutting machine has recently been installed in a slab-mill run-out table in steel works in the United States, to cut hot slabs up to 74 inches in width, and is suitable for the full range of thicknesses from the mill. The machine employs four torches mounted in pairs, one above the other, each pair moving from the outside edges inwards and controlled by an operator from a point conveniently situated near the roller train.

Capital costs of an oxygen cutting machine compare favourably with the cost of an equivalent mechanical or hydraulic shear. Oxygen consumption is naturally dependent on cutting speeds.

The cost of oxygen must, of course, be compared with the cost of power required to operate conventional shears, but it may well be that, in the case of steel works where the installation of a tonnage oxygen plant for other purposes has been decided on, it will be an economic proposition to consider oxygen cutting of hot slabs and blooms and thereby have an additional oxygen demand, sufficient to warrant the installation of a larger plant. This may result in an overall reduction in oxygen costs.

U. S. A.

The Prevention of Oil Film Whirl in Journal Bearings

By R. S. SHERWOOD. (From *Machine Design*, Vol. 25, No. 12, December, 1953, pp. 163-168, 7 illustrations.)

A rational approach to minimising oil film whirl in journal bearings can eliminate some of the costly effort required when the problem is encountered for the first time. The procedures presented in this article for solving the problem of oil film whirl may also serve to forewarn the designer of potential whirl and to indicate methods for steering bearing design away from danger areas.

OIL film whirl in plain journal bearings is a condition in which the centre of the journal travels in an enclosed path about the centre of the bearing in the direction of journal rotation, as shown in Fig. 1. When this condition exists, frequency of whirl is normally something less than one-half journal rotational frequency and occurs at a definite frequency—not erratically. Furthermore, oil film whirl seems to occur largely in lightly loaded journal bearings. However, it does not follow that all lightly loaded journal bearings will be afflicted with oil film whirl. Some outside exciting force seems required to start this whirling action.

In this connection, other forms of journal instability sometimes produce effects similar to oil film whirl, and the following are worth mentioning :—

Shaft Deformation: If a shaft at rest, supported by clearance journal bearings, is heavy enough to have substantial elastic deformation, the centre of the journal will not necessarily coincide with the centre of the bearing. Rotation of this shaft occurs about its elastic axis rather than a line through the centres of the bearings. Thus, if the shaft passes through a critical speed, the centre of the journal may rotate in a closed path about the bearing centre, but at the same frequency as the journal. This is not oil film whirl.

Hysteresis: Newkirk[1] has shown that a journal can whip because of internal friction or hysteresis in the material of the rotating element. Internal friction of a material subjected to a cyclically varying load produces a condition in which stress is not zero when strain is zero—also the converse. The result of this hysteresis is given by Shaw and Macks.[2] Frequency of this type of shaft whirling will usually be very near the natural frequency of the rotor or shaft. However, the shaft can whirl at all rotational speeds above the critical speed, if sufficient hysteresis is present in the material. Thus, this type of whirl is different from the oil film whirl discussed here.

Dry Friction Whirl: Still another form of shaft whirl can exist, either as the journal starts initial rotation or under action of heavy loading. This condition takes place when metal-to-metal contact occurs between the journal and the bearing, and the journal rolls on the bearing. Frequency of this type of whirl can be vastly greater than the rotational frequency of the shaft. If such a condition exists for sustained periods, enough heat energy can be released to cause complete bearing failure. As an additional distinction from oil film whirl, the direction of dry friction whirl is opposite to that of shaft rotation.

CAUSES OF OIL FILM WHIRL

Although oil film whirl actually develops within a plain bearing, its symptoms and adverse effects are experienced in the resonant response of the rotor, a condition which usually cannot be tolerated. For example, a rotor carrying a gear supported on plain bearings may vibrate so badly during oil film whirl that an immediate change of rotational speed is required.

Although a lightly loaded high-speed journal bearing may seem to develop oil film whirl without any external exciting force, the exciting force appears to be necessary. A thick-film lubricated bearing has potential instability, which can sustain oil film whirl once the centre of the journal begins vibrating. Nevertheless, the same bearing, if not disturbed, may operate with

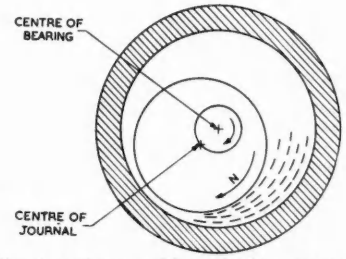

Fig. 1. **Rotation of centre of journal with respect to centre of bearing.**

the centres of the journal and bearing at a fixed relative position, and no oil film whirl develops.

Studies[3, 4] of the action of thick-film journal bearings show that maximum frequency of oil film whirl in an unloaded ideal journal bearing (no end leakage) is one-half the journal rotational frequency. Shaw and Macks[2] have established that the general hydrodynamic equations for a dynamically loaded sleeve bearing of infinite length (neglecting friction forces) can be satisfied with either of the following conditions :

(1) Centre of the journal stationary with respect to the centre of the bearing.
(2) Centre of the journal rotating with respect to the centre of the bearing at a maximum frequency of one-half shaft rotational frequency.

An outside force is required to start oil film whirl. However, once it begins, it will sustain itself by action of the oil film alone, unless there is sufficient damping within the bearing to bring the journal centre to an equilibrium condition. Furthermore, if the lowest resonant rotor frequency is less than one-half shaft rotational frequency, oil film whirl may possibly be sustained from this continuous exciting force.

In a real journal bearing (with end leakage and some load), frequency of oil film whirl is generally considered less than one half the shaft rotational frequency. Cases have been reported in which the frequency of oil film whirl in actual bearings was greater than one-half shaft rotational frequency.[5] However, in view of the limiting frequency established from fundamental considerations of ideal journal bearings, what was called oil film whirl may not have been that alone.

THEORETICAL CONSIDERATIONS

When oil film whirl is encountered in a bearing for the first time, the urgent need for a practical remedy may lead to indiscriminate use of the trial-and-error method. If several factors are changed at the same time, causes and effects are often difficult to set in proper relation.

Bearing eccentricity ratio is a characteristic which serves as an excellent vehicle for theoretical study of variables influencing oil film whirl. Eccentricity ratio for any given journal and bearing varies exactly as the distance between journal and bearing centres changes. In addition, there is a relationship between ratio of oil film whirl frequency to shaft rotational frequency and the eccentricity ratio.

The results of work done by Swift[6] on variation of ratio of oil film whirl frequency N_w to shaft rotational frequency N, as the eccentricity ratio changes, are

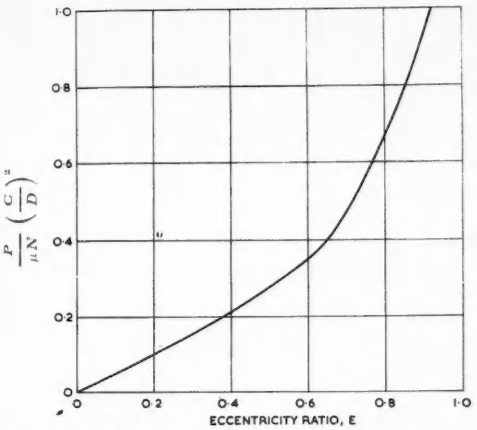

Fig. 2. **Variation of whirl to shaft-rotational frequency with changes in eccentricity ratio.**

shown in Fig. 2, for an ideal bearing of infinite length. In an actual bearing, the ratio of N_w/N would be lower at each eccentricity ratio than is shown by the figure; in reality, the lowest value of N_w/N would be more nearly 0·3 than 0·4.

A most desirable condition prevails when N_w/N is as low as possible, simply because severity of oil film whirl will be minimised if it does occur. To achieve the lowest value, Fig. 2 indicates a preference for larger values of eccentricity ratio E.

The question may be asked whether it is possible to change the eccentricity ratio enough to have real beneficial effect on oil film whirl frequency. Assume a reasonable possibility of increasing E from 0·4 to 0·6 in Fig. 2. At $E = 0{\cdot}4$, $N_w/N = 0{\cdot}45$; at $E = 0{\cdot}6$, $N_w/N = 0{\cdot}42$. If the shaft has the same rotational frequency in either case, the higher eccentricity ratio produces an oil film whirl frequency about 7 per cent lower than at the smaller eccentricity ratio. This is worth attaining, because frequently the speed at which a shaft may whirl is a matter of 10 per cent higher than the speed at which whirl disappears.

At this point, the variables having an effect on eccentricity ratio E must be considered. Eccentricity ratio is a function of bearing pressure P, viscosity μ, speed N, clearance C, and journal diameter D. As a dimensional equality, this relationship can be expressed as

$$E = \phi\left[\frac{P}{\mu N}\left(\frac{C}{D}\right)^2\right] \quad .. \quad .. \quad (1)$$

The arrangement of the several variables in eq. (1) is clearly presented by Faires.[7] Fig. 3, taken from work by Howarth,[8] indicates the numerical relationship between E and the dimensionless group for an ideal full bearing.

Preference is for larger values of eccentricity ratio E, as outlined previously; results from changes in P, μ, N, C, and D can be evaluated from Fig. 3. Also, eccentricity ratio can be expressed as:

$$E = 1 - 2\,h_0/C \quad .. \quad .. \quad (2)$$

where h_0 is the oil film thickness.

Bearing Pressure Variation: Inspection of the dimensionless group in eq. (1) indicates that increasing P is a step in the right direction, as it gives not only the preferred higher values of E (see Fig. 3), but the lower and better values of N_w/N. Furthermore, it can be shown that, as P increases (μ and N being constant), h_0 decreases and so, too, does friction loss—a desirable condition. Also, inspection of eq. (2) shows that a decrease in h_0 results in a larger eccentricity ratio E, if clearance C is constant. Hence, this reasoning simply confirms that, if oil film whirl occurs in an actual situation, it can be minimised by increasing bearing pressure.

Oil Viscosity Variation: Consideration of Fig. 3 shows that, as μ decreases, a larger eccentricity ratio results which, in turn, leads to more desirable lower values of N_w/N. From eq. (2), the effect of decreasing oil film thickness h_0 is to increase eccentricity ratio, and this is helpful in relieving oil film whirl. Increasing oil temperature, thus decreasing viscosity, might also be a help in reducing oil film whirl in certain circumstances.

Shaft Speed Variation: If possible, journal speed N should be decreased when oil film whirl is encountered. Decreasing N also leads to larger eccentricity ratio and to preferred lower N_w/N values.

Clearance Variation: Fig. 3 suggests that increasing clearance might be helpful as a means of obtaining a more favourable N_w/N value by increasing eccentricity ratio. However, eq. (2) shows that what happens to h_0 and bearing stability, when both E and C are changed, is not so clear, particularly when the rates at which C and E change are different. Although several numerical examples worked out by the writer yield smaller h_0 values for an increase in clearance C, it appears wise to compute the effect on h_0 when considering an increase in clearance. By a different approach, Hagg[9] comes to a similar conclusion regarding the effect of varying clearance. In general, increasing values of C/D are attended by a decrease in friction loss and film thickness.

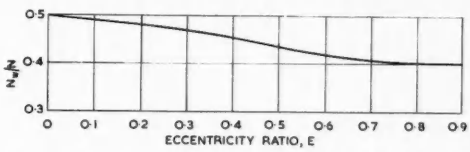

Fig. 3. **Eccentricity ratio relation for an ideal full bearing.**

Bearing Diameter Variation: Effects of a variation in bearing diameter D can be explored in several ways. If other factors are constant, decreasing diameter increases bearing pressure, and this may be considered desirable. It would appear that decreasing diameter would have beneficial effects on friction loss, as well as resulting in a smaller h_0 value.

Practical Remedies

In practice, the problem of oil film whirl in a journal bearing may arise either during the design stage or after the bearing is in service. In the former case, the emphasis is on prevention; the latter requires remedial action. The design engineer has to be able to prevent, as well as to remedy, oil film whirl. These two approaches can be considered separately.

Preventing Oil Film Whirl During Design: In the design stage, it is desirable to recognise whether a journal bearing may be susceptible to oil film whirl in order that preventive action can be taken.

Hagg and Warner[10] present a criterion for shaft stability, which now has the increased validity of an estensive experimental programme. Previously, this criterion had a largely mathematical basis only. Basically, these authors have established that zones of stability and instability, as shown in Fig. 4, can be outlined for particular bearing configurations on charts having dimensionless abscissae $(\mu N/P)\ (R/C)^2$ and ordinates CN^2/g by lines representing various values of the rotor parameter CN_c^2/g, where R is the radius of the bearing and N_c the lowest critical rotor speed.

In practice, CN_c^2/g is computed for the shaft to determine the particular rotor parameter line. Evaluation of CN^2/g and $(\mu N/P)\ (R/C)^2$ for the actual bearing then establishes a point on the chart. If this point is

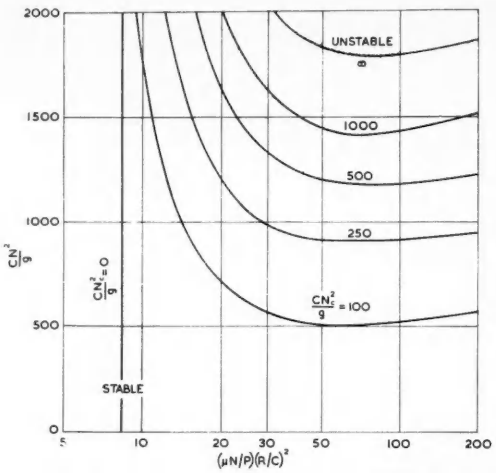

Fig. 4. **Chart for determining zones of stability and instability for 120-deg. centrally loaded clearance bearings with a length-diameter ratio of unity.**

below the line, the bearing design is in a stable region. Several other stability charts are also presented for other bearing length/diameter ratios and load-carrying angles. Another stability criterion is in terms of eccentricity ratio and related attitude angle.[11]

With regard to lowest resonant rotor frequency and shaft rotational frequency, both theory and practice suggest that the lowest resonant shaft frequency should be more than half (say 0·6) of the rotational frequency. This will not always prevent oil film whirl, but it will always minimise the disturbance. Rotor stiffness should be adjusted to give a suitable relationship between lowest resonant frequency and rotational frequency. In this respect, however, it should be mentioned that large power-station generators often have rotor critical speeds less than one-half running speed, with rare incidence of oil film whirl.

Shaft surface speeds above 100 fps have been considered high speed, and oil film whirl frequently has been observed above this value. However, surface speed really is not a suitable criterion ; neither is shaft speed alone, though shaft speed and bearing pressure, considered together, are of interest. Newkirk[12] indicates some results where a 2-in. diameter shaft experienced oil film whirl at 22,000 rpm and 133 psi bearing pressure. Yet, this bearing, after being shortened, operated at 30,000 rpm at a bearing pressure of 266 psi without oil film whirl. These numerical values must not be given undue significance, for a bearing with much lower pressures (or even no pressure) might whirl at much lower speeds, particularly vertical bearings with small radial loads.

Whenever the design engineer gets out of a speed and bearing pressure range known to be safe, he may have to hedge against possible oil film whirl difficulties. One of the safest ways is to design in such a manner that the bearing pressure can be increased, if required. Frequently, this means being able to shorten the bearing.

Redesigning to Eliminate Whirl: If true oil film whirl is present, a reed-type tachometer will give two readings, one at the shaft speed and the other at less than half the shaft rotational speed. Such information is usually good evidence of oil film whirl. Fortunately, a number of practical remedies for such designs exist, and these are worthy of attention.

Increasing bearing oil film pressure is one way to eliminate oil film whirl. Only two methods of increasing bearing pressure will be considered. One involves reducing the load-carrying area of the bearing ; the other involves loading the top of the bearing to increase downward pressure on the journal.

The load-carrying area of a bearing can be reduced by shortening its length, decreasing its diameter, or increasing the size of the chamfer at the bearing split. Frequently, the shaft diameter cannot be changed, because it may already be established from strength or critical speed conditions. Also, changing the shaft size may be too costly.

Shortening the length of a bearing is practical and relatively easily effected. Reducing the length of a bearing has more effect on increasing the bearing oil film pressure than might be expected at first glance. In bearings where L/D is about 1, for example, further reduction of length appreciably increases bearing oil film pressure, because the effective load-carrying area is abnormally influenced by end leakage.

Three variables, which from theory have beneficial effects on eccentricity ratio, also have some practical significance. These are bearing pressure, viscosity, and bearing clearance. Sometimes, oil viscosity depends on outside conditions, such as when the same oil is used to lubricate the gear teeth and the shaft bearings of a speed reducer. On many installations, however, it is perfectly possible to experiment with viscosity changes as a means of eliminating oil film whirl. In fact, experimentation may be the only way to determine accurately the effect of viscosity change on oil film whirl ; existing theory and practice do not always agree on the end result of a viscosity change. Theory suggests reducing viscosity if oil film whirl exists, accomplished in practice either by putting in a lower viscosity oil or by running the existing oil at a higher temperature. If the oil supply system has a heat exchanger, this may be regulated to permit a higher operating oil temperature. As always, proper judgment must be applied to such a remedy. Reducing viscosity reduces film thickness, and this is desirable up to a point. The film must be thick enough to pass foreign matter in the oil safely, for example.

From a practical viewpoint, it would not be too difficult to change the clearance of a bearing operating with oil film whirl. The question is whether to increase or decrease the clearance for beneficial results. It is perfectly possible to show a desirable effect from increasing clearance on eccentricity ratio. However, experimental verification of the end results of a clearance change is lacking, so that it is only prudent to proceed carefully.

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Graphical and Numerical Investigation of Forced Vibrations of a Nine-Mass System

By W. KURZEMANN.

This article describes in detail the application of the vector-diagram method to a damped nine-mass system. As non-dimensional ratios are used, the evaluation applies to any other equivalent system having the same parameters.*

EQUIVALENT LINEAR SYSTEM.

The nine-mass system chosen as an example is the torsional system shown in Fig. 1. It consists of moments of inertia $\mathcal{J}$ [kg cm sec^2] connected together by shafts of stiffness k_t [kg cm/rad], with damping represented by a torsional damping coefficient c_t [kg cm sec/rad]. Under the influence of a sinusoidally-varying torque T [kg cm], each of the masses vibrates with a vibration amplitude θ [rad].

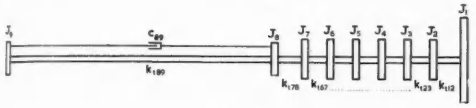


Fig. 1

The system is reduced to a linear system by introducing a reference radius r_{red} [cm] and making use of the following relations :—

Mass :	$m = \mathcal{J}/r^2_{red}$	[kg sec^2/cm]
Force :	$P = T/r_{red}$	[kg]
Displacement :	$a = r_{red}$	[cm]
Stiffness :	$k = k_t/r^2_{red}$	[kg/cm]
Damping coefficient :	$c = c_t/r^2_{red}$	[kg sec/cm].

Non-dimensional quantities are obtained by dividing the amplitudes by a reference value, e.g., a_1, and the forces by $k_{12}\,a_1$. In this way, the forces related to the n-th mass, i.e., the inertia force X_n, the spring force $S_{nn'}$ and the damping force $D_{nn'}$ (where $n' = n + 1$) are expressed as

$$X_n = m_n\,a_n\,\omega^2/k_{12}\,a_1,\quad S_{nn'} = s_{nn'}\,k_{nn'}/k_{12}\,a_1$$
$$\text{and}\quad D_{nn'} = c_{nn'}\,s_{nn'}\,\omega/k_{12}\,a_1$$

ω being the phase velocity of the forced vibration [rad/sec] and $s_{nn'} = a_n - a_{n'}$ = deflection of spring between the masses m_n and $m_{n'}$.

Let the linear system equivalent to that in Fig. 1 be characterised by the following values :—

$m_2/m_1 = \ldots = m_7/m_1 = 0{\cdot}1043$; $m_8/m_1 = 0{\cdot}0467$; $m_9/m_1 = 0{\cdot}0164$; $(m_8 + m_9)/m_1 = 0{\cdot}1207$; $k_{12}/k_{23} = \ldots = k_{12}/k_{67} = 1{\cdot}258$; $k_{12}/k_{78} = 1{\cdot}458$; $k_{89}/k_{12} = 0{\cdot}05117$; and $\psi = 0{\cdot}5$ (rubber damping), so that $c_{89}\,\omega_n/k_{89} = \psi/2\,\pi$ can be subsequently determined, ω_n being the natural phase velocity of the vibrating system. Moreover, use will be made of the quantity

$$\sqrt{k_{12}/m_1} = \omega_1 = 786{\cdot}6 \quad [\text{rad/sec}],$$

which is the natural phase velocity of the sub-system, consisting of stiffness k_{12} and mass m_1.

Finally, it should be noted that the quantity

$$\nu_1^2 = \omega^2/\omega_1^2$$

is the ratio of the forced phase velocity ω to ω_1, squared. In non-dimensional units, it also represents the inertia force X_1, as well as the relative deflection s_{12}/a_1 of the spring between masses m_1 and m_2, since

$$X_1 = m_1\,\omega^2\,a_1/k_{12}\,a_1 = \nu_1^2,\quad \text{and}$$
$$s_{12}/a_1 = s_{12}\,k_{12}/k_{12}\,a_1 = m_1\,\omega^2\,a_1/k_{12}\,a_1 = \nu_1^2.$$

EVALUATIONS WITH ZERO DAMPING AND INFINITE DAMPING.

In describing the method, it is convenient to consider first a force P acting on mass m_8 and then a force P acting on any mass of the system.

Basic system (masses m_1 to m_8) with zero damping : In a system without damping, all the masses vibrate either in phase or in anti-phase, so that the vibration vectors are situated on a straight line. The graphical construction can then be replaced by a numerical evaluation. The forced-frequency tabulation is carried out as indicated in the following table, which is based on a forced frequency corresponding to a phase velocity ω of 500 rad/sec, and an undamped system (masses m_1 to m_8) with force P acting on mass m_8.

The dynamic magnifier M is determined from the relations :—

$$P/k_{12}\,a_1 = -\Sigma X = -0{\cdot}1974\ ;$$
$$M = |\,a_1/(P/k_{12})\,| = 1/0{\cdot}1974 = 5{\cdot}06,$$

and the phase angle between P and a_8 is $\varphi = 0$ deg.

The tabulation is repeated for various ω-values and by plotting the curve of the reciprocals of the dynamic magnifiers against ω (see curves I', I'' in Fig. 2) ; the natural frequency of this system is obtained as the intersection of the curve $1/M$ with the horizontal axis, i.e., for $1/M = 0$. The result for the present numerical example is

$$(\text{system } m_1 \text{ to } m_8) \quad \omega_n = 623 \text{ rad/sec.}$$

Complete system (including mass m_9) with infinite damping : Infinite damping $c_{89} = \infty$ between masses m_8 and m_9 means that these two masses are rigidly coupled together. Thus, at the position of the 8th mass, the system has a combined mass $m_8 + m_9$ (including the mass of the shafting).

*See *The Engineers' Digest*, March 1953, pp. 78-80, and November, 1953, pp. 421-423.

$$\omega = 500 \text{ rad/sec} ;\ \nu_1^2 = \omega^2\,m_1\,a_1/k_{12}\,a_1 = (500/786{\cdot}6)^2 = 0{\cdot}404 ;\ \nu_1 = 0{\cdot}636.$$

Mass No.	(1) Deflection $\frac{a_n}{a_1} = \frac{a_{n-1}}{a_1} - \frac{s_{nn'}}{a_1}$	(2) Mass ratio $\frac{m_n}{m_1}$	(3) Force $X_n = \frac{m_n}{m_1} \times \frac{a_n}{a_1} \times \nu_1^2$	(4) Total force ΣX	(5) Spring ratio $\frac{k_{nn'}}{k_{12}}$	(6) Spring deflection $s_{nn'} = (\Sigma X) \times \left(\frac{k_{nn'}}{k_{12}}\right)$
1	1·0000	1·0000	0·4040	0·4040	1·0000	0·4040
2	0·5960	0·1043	0·0251	0·4291	1·2580	0·5400
3	0·0560	0·1043	0·0024	0·4315	1·2580	0·5425
4	−0·4865	0·1043	−0·0205	0·4110	1·2580	0·5170
5	−1·0035	0·1043	−0·0423	0·3687	1·2580	0·4640
6	−1·4675	0·1043	−0·0618	0·3069	1·2580	0·3860
7	−1·8535	0·1043	−0·0782	0·2287	1·2580	0·3322
8	−2·1857	0·0467	−0·0413	0·1974	—	—

Note : a_2/a_1 = amplitude of mass m_2, obtained by subtracting in previous line the value of Col. (6) from that of Col. (1). Similar procedure for a_3/a_1, etc.

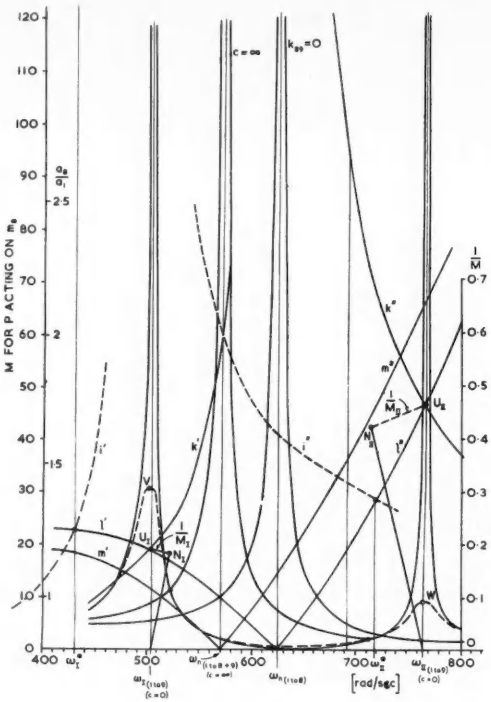


Fig. 2

The forced-frequency tables are the same as in the previous case, excepting that $(m_8 + m_9)/m_1 = 0{\cdot}1207$ is used in place of $m_8/m_1 = 0{\cdot}0467$. For $\omega = 500$ rad/sec and $\nu_1^2 = 0{\cdot}404$, the table ends as follows :—

Mass No.	(1) $\frac{a_n}{a_1}$ = (1) — (6) of previous line	(2) $\frac{m_n}{m_1}$	(3) X_n = (1) × (2) × ν_1^2	(4) ΣX	(5) $\frac{k_{nn'}}{k_{12}}$	(6) $s_{nn'}$ = (4) × (5)
8	−2·1857	0·1207	−0·1067	0·1220	—	—

Therefore, $P/k_{12}\ a_1 = -\Sigma X = -0{\cdot}1220$; $M = |\ a_1/(P/k_{12})\ | = 1/0{\cdot}1220 = 8{\cdot}2$; and $\varphi = 0$ deg. (phase-angle between P and a_8).

By determining M for various ω-values, the curve of $1/M$ can be plotted against ω (curves m', m'' in Fig. 2) and its intersection with the horizontal axis ($1/M = 0$) gives the natural frequency of this system, *viz.* :

(system m_1 to $m_8 + m_9$) $\quad \omega_n = 570$ rad/sec.

Complete system (including mass m_9) with zero damping : From the natural phase velocities of the basic system (masses m_1 to m_8) and the damper system (mass m_9 and shaft k_{89}) the "tuning ratio" of the damper system is obtained :

$$\sigma = \omega_9/\omega_{n(1\text{-}8)} = \sqrt{k_{89}/m_9}/623.$$

Moreover, the forced phase velocity ω divided by ω_9 gives the "frequency ratio" $\nu_9 = \omega/\omega_9$.

It is convenient to begin the evaluation from mass m_9, assuming a_9 is known. The value of a_8 is then obtained from

$$s_{98}/a_9 = (a_9 - a_8)/a_9 = s_{98}\ k_{89}/a_9\ k_{89} = m_9\ \omega^2\ a_9/k_{89}\ a_9 = \nu_9^2 = X_9' \text{ so that } a_8/a_9 = 1 - \nu_9^2.$$

The basic system m_1 to m_8 being already evaluated, it is necessary to relate the above equations based on the deflection a_9 to the deflection a_1. This is achieved as follows, for $\omega = 500$ rad/sec :

$\sum_1^8 X = 0{\cdot}1974$ and $a_8/a_1 = -2{\cdot}1857$ (see frequency table);

for $\sigma = 1$: $\nu_9^2 = (\omega/\omega_9)^2 = (\omega/\sigma\ \omega_n)^2 = (500/623)^2 = 0{\cdot}625$.

Hence, $a_8/a_9 = 1 - \nu_9^2 = 0{\cdot}375$. Furthermore, $a_8 = -2{\cdot}1875\ a_1 = 0{\cdot}375\ a_9$, so that $a_9/a_1 = 5{\cdot}84$ and $X_9 = X_9'\ k_{89}\ a_9/k_{12}\ a_1 = 0{\cdot}625 \times 0{\cdot}05117 \times 0{\cdot}625 = 0{\cdot}187$.

X_9 is in the same direction as a_8/a_9. The equilibrium of forces acting at mass m_9 gives the relation

$$-P/k_{12}\ a_1 = \sum_1^8 X - X_9 = 0{\cdot}1974 - 0{\cdot}187 = 0{\cdot}01\ ;$$

hence, $M = 100$.

This calculation can be repeated for various ω-values to obtain the natural frequency from $1/M = 0$. As, however, in Fig. 2 the ordinates of the curves l' and l'' represent $\sum_1^8 X$, it can be seen that, for a graphical determination of the resultant M-curve, all that is necessary is to plot X_9 against ω (curves k' and k''). The difference in the ordinates of k' and l' (or k'' and l'') is equal to $1/M$ for any ω-value. Hence, their points of intersection U_I and U_{II} correspond to $1/M = 0$ and give the two (lowest) natural phase velocities of the undamped system : $\omega_I = 503$ rad/sec and $\omega_{II} = 764$ rad/sec.

For another value of the tuning ratio, *viz.*, $\sigma^* = \omega_9/\omega_{n(1\text{-}8)} = 0{\cdot}643$, the curves i' and i'' are obtained for X_9 plotted against ω. The intersections with the curves l' and l'' give the corresponding one-node and two-node natural phase velocities : $\omega_I^* = 430$ rad/sec and $\omega_{II}^* = 718$ rad/sec (see Fig. 2).

Force Acting on any Mass.

As, in engines, the harmonics of the excitation forces can be regarded as acting at the centre of each crankthrow, it is necessary to determine the effects of a force P acting on any of the engine masses.

Basic system (masses m_1 to m_8) with zero damping : The forced frequency tables of this system, with the evaluation beginning from mass m_1, have already been calculated.

A second set of tables beginning with mass m_8 is prepared, with vibration amplitudes based on a_8. The two sets of tables are then correlated to each other by means of the line corresponding to the mass on which the force is acting. For instance, if $P = P_6$ is a force acting on mass m_6, the relation is

$$(X_1 + \ldots + X_5) + X_6 + (X_7' + X_8') = P_6$$

where the X-values are based on a_1 and the X'-values on a_8.

Thus, for a forced phase velocity $\omega = 550$ rad/sec, or $\nu^2{}_{(1\text{ to }8)} = (\omega/\omega_n)^2 = (550/632)^2 = 0{\cdot}755$ and a value $\omega_8 = \sqrt{k_{78}/m_8} = 3295$ rad/sec, the conversion is as follows :

$a_6 = 1{\cdot}868\ a_1 = 0{\cdot}8861\ a_8$, so that $a_8/a_1 = 2{\cdot}106$;

$$X_7 + X_8 = (X_7' + X_8')\ \frac{k_{78}\ a_8}{k_{12}\ a_1} = 0{\cdot}096 \times 0{\cdot}688 \times 2{\cdot}106 = 0{\cdot}1442.$$

Hence, $-P_6/k_{12}\,a_1 = 1/M = X_1 + X_2 = \ldots + X_8 = 0{\cdot}3033 - 0{\cdot}1442 = 0{\cdot}1592$ and $M = 1/0{\cdot}1592 = 6{\cdot}3$ is the corresponding dynamic magnifier of the system.

Similar calculations for a force P acting on different masses give the following results :—

P acting on mass	m_1	m_2	m_3	m_4	m_5	m_6	m_7	m_8
$M = a_1/(P/k_{12})$	0·81	0·57	2·22	3·92	5·35	6·30	6·97	7·30
a_8/a_9	8·32	6·4	0·421	1·405	1·839	2·106	2·315	2·523

This shows that high values of the resonance function are obtained when the mass on which P acts is remote from the node of the vibrating system.

Complete system (including mass m_9) with zero damping : The forced-frequency tables of the 9-mass system are obtained from those of the 8-mass system beginning with mass m_1 by extending these tables so as to include m_9.

Furthermore, tabulations beginning from mass m_9 and vibration amplitude a_9 are carried out. The correlation of the two sets of tables for each mass on which the force P acts is obtained as in the previous case for the 8-mass system. The results for the numerical example are as follows :—

P acting on mass	m_1	m_2	m_3	m_4	m_5	m_6	m_7	m_8	m_9
$M = a_1/(P/k_{12})$	6·67	4·78	2·09	0·73	3·45	5·95	8·06	10·1	30·13

High M-values are again reached when the force P is acting on masses remote from the node of the vibration. In this connection, it is useful to draw the mass-elastic line of the system for each case, i.e., the curve of the vibration amplitudes of the various masses plotted against equivalent shaft lengths.

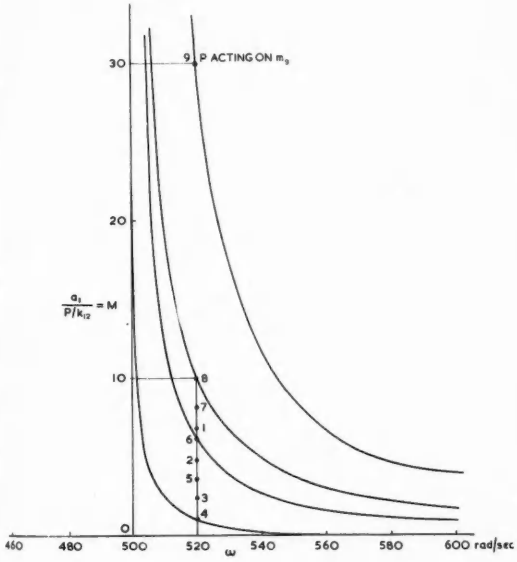


Fig. 3.

Finally, it is of interest to consider Fig. 3, which shows that the resonance curve with zero damping is appreciably different according to the position, relative to the node, of the mass on which the force P is applied.

EVALUATION OF RESONANCE CURVES WITH FINITE DAMPING.

Force P acting on mass m_1 or m_8 : The resonance function for the damped 9-mass system can be obtained by the method used for determining the curves in Fig. 2 for this same system without damping. In other words, the basic system (m_1 to m_8) is combined with the auxiliary system (k_{89} and m_9) to obtain results for the complete 9-mass system.

It is now necessary, however, to construct vector diagrams for the forces and vibration amplitudes, since their phase angles are no longer zero or 180 deg. The vector diagram of the damper mass m_9 is constructed as shown in Fig. 4a (for $\omega = 500$ rad/sec) :—

(1) From point O_9', draw a vector $\mathbf{a}_9$ of unit length $\overrightarrow{O_9'A'}$. Along this vector, from O_9', mark off the length $v_9^2 = \omega^2/(k_{89}/m_9) = s_{98}/a_9 = X_9' = \overrightarrow{O_9'B'}$.

(2) Determine the phase angle of damping, δ_{89}, from $\tan \delta_{89} = c_{89}\,\omega/k_{89} = \psi/2\pi$ (with $\psi = 0{\cdot}5$ for rubber) and draw from O_9' a line forming an angle δ_{89} with X_9'. From the point B', erect a perpendicular to this line, intersecting it at point C'.

(3) Draw the vector $\overrightarrow{A'C'} = -\mathbf{a}_8/a_9$, this completing the triangle.

These vectors must now be related to those of the basic system. From the forced-frequency table for the complete system with zero damping, the length of $\mathbf{a}_9/a_1$ is known, *viz.* : $|\mathbf{a}_8/a_1| = -2{\cdot}1857$, and $\mathbf{a}_8/a_1$ is in the same direction as $+\mathbf{a}_9/a_9$. Moreover, Fig. 4a gives the length of $a_8/a_9 = 0{\cdot}381$. Therefore, $a_8 = 0{\cdot}381\,a_9 = -2{\cdot}1875\,a_1$, and $a_9/a_1 = -5{\cdot}7404$.

This procedure should be repeated for both natural phase velocities ω_I and ω_{II} of the 9-mass system, i.e., for $v_9^{2}{}_{(I)} = (500/623)^2 = 0{\cdot}625$ and $v_9^{2}{}_{(II)} = (764/623)^2 = 1{\cdot}501$ (vectors $\overrightarrow{O_9'B''}$ and $\overrightarrow{A'C''}$). The corresponding resonance curves can be determined in Fig. 2, using the diagrams in Figs. 4b and 4c, which are constructed as follows (for $\nu_9 = \nu_{9(I)}$) :—

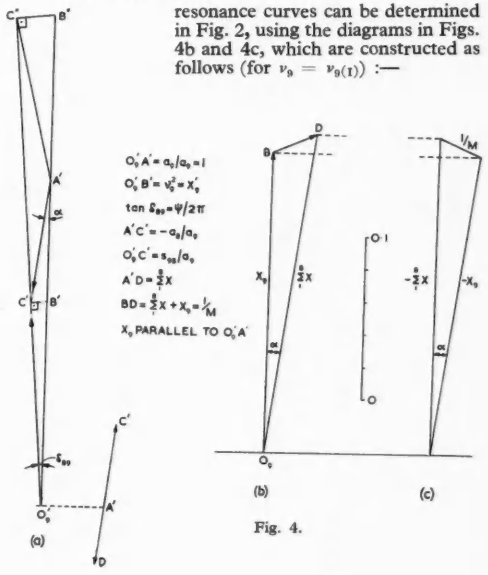


Fig. 4.

(1) Calculate $X_9 = X_9' \times (k_{89}/k_{12}) \times (a_9/a_1) = 0{\cdot}625 \times 0{\cdot}0512 \times (-5{\cdot}7404) = -0{\cdot}184$ and plot this vector (in the direction of $O_9'B'$) in the diagram of Fig. 4b.

(2) From O_9 plot the vector $\sum_1^8 X = -O_9D$ (taking its numerical value from the frequency tabulation), using the same scale as for X_9, with an angle α, determined from Fig. 4a, between these two vectors. Then $DB = 1/M$, where M is the dynamic magnifier of the complete system with damping.

Fig. 4c shows that $1/M$ can also be obtained by erecting first a vector $-\sum_1^8 X$ and then, at an angle α to it, the vector $-X_9$. This procedure is more convenient for direct plotting of the resonance curve in Fig. 2.

In Fig. 2, the ordinate joining the point ω_I to U_I (on curve l') gives $\sum_1^8 X$ at resonance. Therefore, the construction of Fig. 4c can be carried out directly in Fig. 2, using the $1/M$-scale on the right-hand side of this figure. The vector $\overrightarrow{\omega_I N_I}$ is equal to $X_9 = 0{\cdot}184$ and $\overline{N_I U_I} = 1/M$. Its reciprocal, M, determines the point V, which is at the peak of the damped resonance curve for $\psi = 0{\cdot}5$.

The procedure is then also applied to the second case, i.e., $\nu_9^2{}_{(II)} = 1{\cdot}501$, which gives

$X_9 = 1{\cdot}507 \times 0{\cdot}0512 \times (2{\cdot}895/0{\cdot}570) = 0{\cdot}436 = \overline{\omega_{II} N_{II}}$

thus determining $1/M$ and hence point U_{II}, the peak of the second resonance curve of the damped system, shown in Fig. 2.

The vector diagram of the 9-mass system with damping can also be constructed with the aid of a table beginning with mass m_9. The details of the procedure, for $\omega = 764$ rad/sec and $m_9/m_9 = 0{\cdot}6641$ (damper mass twice as large as in the previous cases) are shown in Fig. 5 and the frequency tabulation given below.

$\omega = 764$ rad/sec.

$m_8/m_9 = 0{\cdot}25$; $m_2/m_9 = \ldots = m_7/m_9 = 0{\cdot}6641$; $m_1/m_9 = 6{\cdot}368$; $k_{89}/k_{78} = 0{\cdot}1431$; $k_{89}/k_{23} = \ldots = k_{89}/k_{67} = 0{\cdot}124$; $k_{89}/k_{12} = 0{\cdot}09859$.

$\omega_{n(1\text{ to }8)} = 623$ rad/sec ; $\sigma = 1$; $\omega_{n9} = 623$ rad/sec ; $\nu_9^2 = 1{\cdot}503$; $\psi_{89} = 0{\cdot}5$; $\tan \delta_{89} = 0{\cdot}5/2\pi = 0{\cdot}0794 = AA'$.

(The lengths refer to the vectors in Fig. 5). Formulae used :

$X_n' = (m_n/m_9) \times (a_n/a_9) \times \nu_9^2$; spring deflection $= s_{n'n}/a_9 = S'_{n'n}\, k_{89}/k_{n'n} = OY' \times k_{89}/k_{n'n}$.

Regarding the construction of Fig. 5, which is built up at the same time as the tabulation is being carried out, it should be noted that : $a_8/a_9 = 08 = AB$ parallel to CD ; OD' is on OD ; DE is parallel to $D'8$; $a_7/a_9 = 07 = D'8$; OE' is on OE ; EF is parallel to $E'7$; $E'7 = 06$; etc.

Mass No.	a_n/a_9 (from Fig. 5)	X_n'	$\Sigma X' +$ Damping force $= Y'$	$s_{n'n}/a_9$
9	$1{\cdot}000 = OA$	$1{\cdot}503 = OC$	$1{\cdot}500 = OB$	$1{\cdot}500 = OB$
8	$0{\cdot}510 = AB$	$0{\cdot}192 = CD$	$1{\cdot}320 = OD$	$0{\cdot}189 = OD'$
7	$0{\cdot}690$	$0{\cdot}689 = DE$	$0{\cdot}660 = OE$	$0{\cdot}0818 = OE'$
6	$0{\cdot}765$	$0{\cdot}764 = EF$	$0{\cdot}225 = OF$	$0{\cdot}0330 = OF'$
5	$0{\cdot}752$	$0{\cdot}752 = FG$	$0{\cdot}917 = OG$	$0{\cdot}1140 = OG'$
4	$0{\cdot}640$	$0{\cdot}639 = GH$	$1{\cdot}540 = OH$	$0{\cdot}1910 = OH'$
3	$0{\cdot}450$	$0{\cdot}450 = HJ$	$1{\cdot}980 = OJ$	$0{\cdot}2455 = OJ'$
2	$0{\cdot}210$	$0{\cdot}210 = JK$	$2{\cdot}175 = OK$	$0{\cdot}2140 = OK'$
1	$0{\cdot}080$	$0{\cdot}767 = KL$	$2{\cdot}125 = OL$	—

$LO = 1/M_9 = (P_1/k_{89})/a_9 = 2{\cdot}125$; $M_9 = 0{\cdot}471$.

Force P acting on any mass : The procedure is the same as in the case where no damping is present. For a value ω of the forced-phase velocity, the table and vector diagram beginning from m_1 are worked out up to the mass on which P acts, for instance m_6 if $P = P_6$. A table and vector diagram beginning from mass m_9 are then constructed up to m_6. The relations a_6/a_1 and a_8/a_9 give a_1/a_9. Finally, the polygon of forces acting on m_6 is determined. It consists of : $X_1' \ldots + X_5'$ (based on $k_{89}\, a_9$ and related to a_6/a_9 with the correct phase angle) ; $X_7' + \ldots + X_9'$; X_6' (parallel to a_6) ; and the excitation force P, which closes the polygon. As $P/k_{89}\, a_9 = 1/M$, its reciprocal gives the dynamic magnifier for the mass considered.

For the present example, with forces acting on $m_2, m_3, \ldots, m_7$, after constructing diagrams according to the above method, the dynamic magnifiers (M_2 to M_7) would be found to have the following values : 0·018, 0·425, 0·827, 1·13, 1·57, and 1·5, respectively.

Thus, the complete contour of the resonance curve of the system with damping can be obtained by means of vector diagrams. The results are on a correct theoretical basis, since the diagrams are exact graphical solutions of the differential equations of the vibrating system under forced-vibration conditions.

APPLICATIONS.

The above method can be applied to any type of vibrating system with damping.

In the case of reciprocating engines, it is necessary to combine the excitation forces according to the phase vectors for the harmonic order number of the vibration considered, taking account of the crank angles and firing sequence.

The resonance amplitude $a_{n(res)}$ of any mass m_n is given by the vector sum*

$$a_{n(res)} = \left[M_{[9]} \frac{a_n}{a_9}(P_2) + M_{[9]} \frac{a_n}{a_9}(P_3) + \ldots + M_{[9]} \frac{a_n}{a_9}(P_7) \right] \times a_{st[9]} = \sum_{(2)}^{(7)} M_{[9]} \frac{a_n}{a_9} \times a_{st[9]}$$

(*Concluded on page* 68)

*The M-values should be drawn along the lines of action of the phase vectors of the star diagrams, but in the negative direction in each case. The index (n) refers to a force P acting on mass m_n, and the index [9] indicates that the corresponding values are based on the use of a_9 in the denominator.

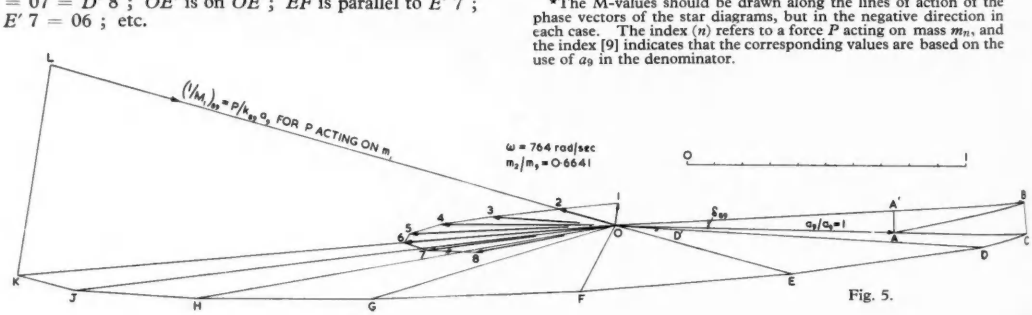


Fig. 5.

Apparatus for Checking Contours of Propeller Models

By P. LANGSETHMO. (From *Teknisk Ukeblad*, Vol. 100, No. 35, September 24, 1953, pp. 763-764, 5 illustrations.)

MODELS of marine propellers are usually checked with an apparatus which determines the pitch and the blade thickness at certain points. It may be desirable, however, to obtain a more extensive check of the correctness of the blade contours. This can be achieved by means of the apparatus described below, which automatically plots out the blade sections on paper.

The main features of the equipment, which includes motors and relays operating on 24 volts d.c., supplied by a transformer and rectifier unit, are shown in Fig. 1. The propeller P and the recording drum T are mounted on a shaft B, which is slowly rotated by the motor M_1 through spur and worm gearing. A feeler F and a stylus S are mounted at opposite ends of a horizontal arm A. This arm can be raised or lowered by rotating a micrometer screw, so as to obtain accurate adjustment for each cylinder-section diameter. The arm A is supported by a frame on the shaft B, and can be displaced axially along the shaft by motor M_2 driving a lead screw. As the feeler and stylus are both rigidly attached to the arm A, the distance between them is always the same, so that the stylus will give an accurate reproduction on the recording drum of the relative motion of the feeler around the propeller.

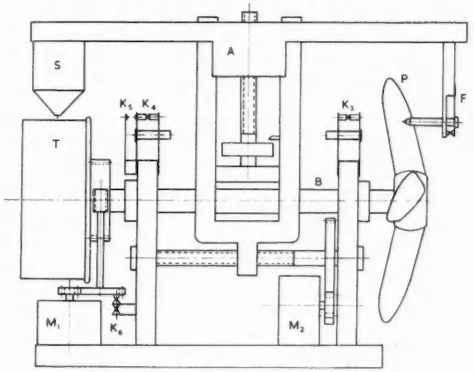


Fig. 1.

The feeler and stylus controls are shown schematically in Fig. 2. The point of the feeler F is mounted on a leaf spring f. When the feeler touches the propeller blade, it closes an electric circuit through the propeller blade, the feeler, and coil L_1 in the quick-acting polarised relay R, which requires very little current, thus avoiding high-intensity sparks which might otherwise damage the point of the feeler and burn holes in the propeller blade surface. When current flows through coil L_1, the relay is triggered and closes contact K_2. As a result of the current passing through the stylus relay, the stylus is pressed downwards on to the recording drum and perforates a small hole in the paper. As the feeler moves further along the propeller blade, spring f is deflected and interrupts the current at one of the contacts K_0 and K_1. In tests, it has been found that the motion of the feeler is less than 0·01 mm during the period of effective contact at K_0 and K_1 and between the feeler point and the propeller blade. As the polarised relay R does not return to its initial position when coil L_1 is not energised, this relay is provided with another coil L_2. The circuit of L_2 is closed by contact K_2 when the stylus is depressed. This causes relay R to return to its starting position, thus breaking the circuit in the stylus relay. A spring returns the stylus to its initial position.

Fig. 2 also shows the simple change-over mechanism

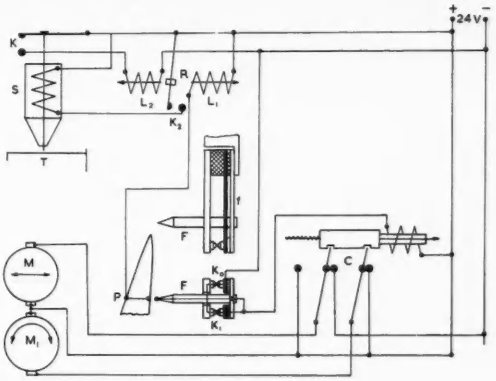


Fig. 2.

between motors M_1 and M_2 necessary for recording the propeller contour. When relay circuit-breaker C, which is controlled by the contacts K_0 and K_1, is without current, it short-circuits motor M_1 and closes the circuit for motor M_2. When current passes through C, it closes the circuit for M_1, and M_2 is short-circuited.

Plotting of the propeller blade contour takes place as follows : Motor M_2 moves the feeler up to the blade surface. At the moment of contact, the circuit is closed by coil L_1. The polarised relay acts and closes the circuit of the stylus relay, and the stylus needle perforates the paper. Contact K above the stylus closes the circuit for coil L_2 and polarised relay R returns to its starting position. M_2 again moves the feeler further inwards on the propeller blade, spring f deflects, and contacts K_1 and K_2 open. The relay circuit-breaker C then operates and short-circuits motor M_2, which stops rapidly, and switches in motor M_1, which rotates the propeller and the recording drum in such a way that the feeler will move a slight distance away from the blade, depending on the pitch of the propeller. As soon as the feeler point leaves the blade, a new current impulse is obtained through the polarised relay, actuating the stylus relay and thus producing a hole in the paper. Motor M_1 is short-circuited and stops rapidly, then motor M_2 starts again, and the same operation is repeated. Thus, the motors are alternately switched in and out of circuit, with M_2 moving the feeler towards the blade and M_1 rotating the blade away from the feeler. Motor speeds can be individually adjusted by means of rheostats, so as to obtain variations in operating speed, and to space perforations as desired on the paper. A set of recording drums of various sizes makes it possible to record true blade contours at various radii. Two feelers, one for the pressure side and one for the suction side of the propeller, can be swung alternately into position.

Contacts K_3 and K_4 switch out motor M_2 when the feeler reaches the end position. When the feeler is at the left end position, in addition to contact K_5, another contact K_6, actuated by the transmission of motor M_1, closes a circuit to coil L_1 in the polarised relay. This occurs once for each complete revolution of the transmission wheel, and results in a recording of a base line, obtained by means of M_1. The contour obtained on the paper is about 6 cm wide and 2·5 cm high, with a base line of perforations some 3 mm apart, and a cylindrical-section blade contour at an angle to the base line, with perforations at intervals between 0·5 and 2 mm. The recording can be magnified and photographed if necessary.

The Balancing of Grinding Wheels in situ

By K. SKRZIVAN. (From *Stanki i Instrument*, No. 8, 1953, pp. 12-17, 26 illustrations.)

GRINDING wheels must be balanced before assembly in the grinding machine, and also after truing of the wheel or other cases of loss of balance. Unbalance may arise from several causes, including uneven wheel wear and non-uniform soaking with coolant (during standstill, the liquid inside the wheel tends to filter into its lower part).

A substantial drawback of the usual method of static balancing is the length of time needed and the relatively short period during which the grinding wheel remains accurately balanced. The dynamic balancing of wide-face wheels outside the grinding machine also takes a long time and can be accomplished only with the help of a balancing machine.

The Czech Institute for Machine Tool and Metal Cutting Research has developed a balancing attachment which enables the balancing of grinding wheels to be carried out *in situ* during the rotation of the grinding wheel. This unit is based on the displacement of masses inside the attachment, which is mounted on the grinding wheel. The simplest design contains balancing masses, which are displaced in relation to the grinding wheel along a spiral, for instance, an archimedean spiral. To simplify manufacture, a sinusoidal spiral has been adopted (Fig. 1). This spiral is, in fact, followed by the common centre T of the two masses Z_1 and Z_2 moving around the grinding wheel at rotational speeds of n_1 and n_2 rpm, having only a small difference between them.

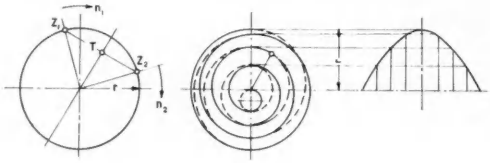


Fig. 1.

The centre of mass of the two masses rotates with angular velocity

$$\omega = (\omega_1 + \omega_2)/2 = \pi (n_1 + n_2)/60$$

and simultaneously moves in the radial direction. As a result of these components of motion, the mass centre describes a sinusoidal spiral, as shown in Fig. 1.

The equation of the spiral, in polar co-ordinates, has the following form :

$$\varphi = r \sin k\omega_T$$

where $$k = (n_1 - n_2)/(n_1 + n_2)$$

In Fig. 2 the position of the balancing attachment on the flange of the grinding wheel is shown. Inside the unit there are two masses, whose displacement in relation to the wheel is controlled during the rotation of the wheel by arresting one or other of the two concentric knobs situated at the face of the unit. When the outer knob is arrested, the masses are displaced at half the speed of, and in the opposite direction to, the displacement caused by arresting the rotation of the inner knob.

The outer knob is intended for the adjustment of the position of the masses and the inner knob for the rapid displacement of the mass centre by several turns of the spiral.

Observation of the presence of vibration is made with the help of a mercury vibroscope. This simple instrument fulfils the requirements of accurate balancing and is based on the observation of the reflection from a mercury surface of a point source of light. If the surface of the mercury is steady, the luminous source is reflected clearly; if, however, the instrument is placed on a vibrating foundation, the mercury surface becomes rippled, and the reflection of the light source is broken up according to the amplitude of vibration. This method does not permit a measurement of the vibration amplitude, but constitutes an extremely sensitive null method of detecting the presence of vibration of an amplitude as low as 0·5 micron.

The accuracy of balancing depends on the pitch of the windings of the spiral. In existing types of small attachments the maximum amount of unbalance which can be compensated is limited to 800 gram-cm, and in large units the maximum unbalance can reach 9000 gram-cm. These limits have been found adequate for grinding wheels of standard quality. If the mean pitch of the spiral constitutes 0·02 of its maximum radius, then the residual unbalance will reach approximately 0·01 of the maximum unbalance which can be handled by the attachment. Consequently, for small units, the unbalance tolerance will be 8 gram-cm, and for large units 90 gram-cm. It has been established by tests that this is sufficient to reduce the amplitude of vibration of the grinding wheel carriage to about 1 micron.

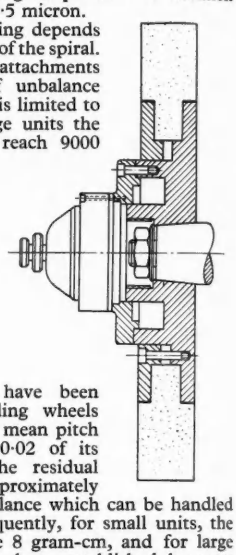


Fig. 2.

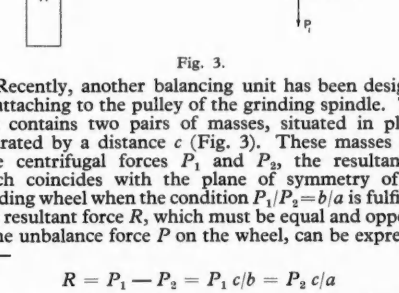


Fig. 3.

Recently, another balancing unit has been designed for attaching to the pulley of the grinding spindle. This unit contains two pairs of masses, situated in planes separated by a distance c (Fig. 3). These masses produce centrifugal forces P_1 and P_2, the resultant of which coincides with the plane of symmetry of the grinding wheel when the condition $P_1/P_2 = b/a$ is fulfilled. The resultant force R, which must be equal and opposite to the unbalance force P on the wheel, can be expressed as :—

$$R = P_1 - P_2 = P_1 c/b = P_2 c/a$$

The mounting of this latest type of balancing unit at the rear of the grinding spindle has the additional advantage that the unit does not have to be removed with each replacement of the grinding wheel.

It is stated that balancing by means of these attachments is achieved very rapidly in practice, and that grinding machine operators do not require special theoretical instruction.

Some Stall and Surge Phenomena in Axial-Flow Compressors

By M. C. HUPPERT and W. A. BENSER. (From *Journal of the Aeronautical Sciences*, Vol. 20, No. 12, December 1953, pp. 835-845, 12 illustrations.)

Observations of rotating stall have shown that a wide variety of stall patterns is possible. Blade failures attributable to resonant vibrations excited by rotating stall have been experienced in single- and multi-stage compressors. This paper discusses the various aspects of stall and surge in axial-flow compressors. Much research must still be done on such problems as correlations of types of stall with stage geometry, evaluations of stage interactions when rotating stall exists in multi-stage compressors, and evaluation of the system characteristics governing surge.

INASMUCH as multi-stage compressor stall is a result of individual stage stall, a study of compressor stall requires an understanding of individual stage characteristics and the stage operating conditions for off-design operation. For this paper the discussion of this phenomenon has been divided into two phases. The first is a study of stall characteristics of multi-stage compressors and the effects of the individual stage characteristics on multi-stage compressor stall. As the experimental technique of studying multi-stage compressor stall may be hampered by surge and stage interactions, it is difficult to evaluate stage characteristics in the proximity of overall compressor stall. Consequently, much of the investigation reported herein has been done by analytical techniques.

STALL OF SINGLE-STAGE AXIAL-FLOW COMPRESSORS

Single-stage stall characteristics must be evaluated in order to understand multi-stage compressor stall characteristics. Performance of a single-stage axial-flow compressor is generally measured with steady-state instrumentation having sufficient damping to attenuate fluctuations with frequencies over 5 or 10 cps. As the flow is decreased beyond the stall point, the flow in all blade passages consequently appears to be uniform with time. With some stages, regions of flow rate exist where surge or low-frequency pulsation is encountered, but steady-state conditions are again indicated if further reductions in flow rate are made. Recent studies with high-frequency-response instrumentation, such as hot-wire anemometers and pressure transducers, have shown stage stall to be a non-steady type of phenomenon, in which some sectors of the annulus are operating at an extremely low mass-flow rate, as compared with the remainder of the annulus; these low-flow sectors rotate about the compressor axis.

At some point far upstream of the stall zone there exists a uniform flow. The stream lines immediately upstream of the blades must, therefore, diverge to maintain continuity around the low-flow or stall zone. This spillage of flow to either side of the stall zones, which tends to increase the angle of attack on one side of the stall and decrease it on the other, results in a tangential propagation of the stall zone. The propagation rate of the stall region relative to the rotor is generally opposite in direction but lower in magnitude than the rotor-blade speed. On the absolute basis, therefore, the stall rotates in the direction of the rotor but at a lower speed. It appears that in principle the rotating or propagating stall may be instigated by a stator as well as a rotor.

To-date, no definite relation between the number of stall regions in the stall pattern and compressor geometry has been determined. Two fairly distinct categories of stall patterns, however, have been observed: one, a progressive stall, originates at the blade ends and consists of multiple stall zones extending over only a portion of the blade span; the other, a root-to-tip stall, generally consists of only one zone extending over the entire blade span.

Progressive Stage Stall: The progressive-type stall has been observed on several low hub-tip ratio stages of the basic solid-body rotation type. For this type of stage operating with uniform inlet flow, the rotor-tip angle of attack varies much more rapidly with flow rate than does the root angle of attack, and the rotor-blade tip naturally stalls first, while the remainder of the blade is inherently unstalled. The circumferential and radial extent of the low-flow zones at the blade tip increases with further decreases in flow rate, while the root-section operating condition remains relatively unchanged. Other types of stage diagram, such as a free-vortex flow, might have a progressive-type stall originating at the root section of the blade. In general, however, jet-engine requirements of high flow capacity and high stage-pressure ratio lead to inlet-stage designs, in which the rate of change of loading with flow at the blade tip is much greater than that at the blade root.

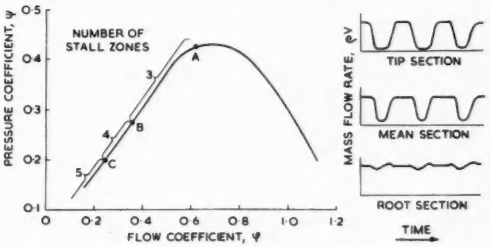


Fig. 1. **Performance of low hub-tip ratio stage with progressive stall.**

A typical performance curve for a stage that is critical at the blade tip is shown in Fig. 1 as a plot of pressure coefficient ψ against flow coefficient φ. As stall first occurred (point *A*), three stall zones were found. The oscillograms of mass-flow rate obtained by hot-wire anemometers for the tip, mean radius, and root section are also shown. The curves of flow rate against time show the greatest flow-rate variations at the tip, a smaller variation at the mean radius section, and only minor flow variations at the root. As the flow is further reduced from the stall point, the area of each stall zone increases. Reductions in flow to point *B* (Fig. 1) resulted in a sudden transition from three to four stall zones, and still further reductions of flow resulted in a transition to five stall zones at point *C*. Thus, the changes in area of individual stall zones and changes in number of stall zones result in a relatively uniform rate of change of stalled flow area with flow rate. Progressive-type stall patterns having from three to eight stall zones with rotational speeds of 40 to 85 per cent of rotor speed have been observed. For each stall pattern, the rotational speed of the stall zones varied directly with rotor speed. It should be noted that progressive-type rotating stall results in a gradual drop in pressure coefficient as the severity of stall increases.

Root-to-Tip Stage Stall: When the stall angle of attack is approached nearly simultaneously over the entire span of the blade, as in high hub-tip ratio stages, the resultant flow-rate variations are relatively uniform in intensity from root to tip. Observations of this type of stall, defined as a root-to-tip stall, show only one

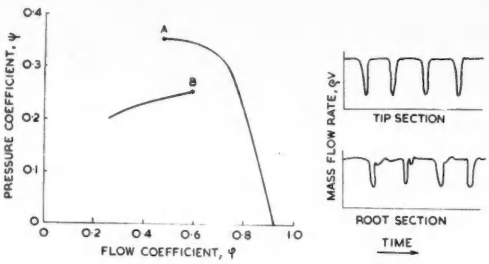


Fig. 2. **Performance of high hub-tip ratio stage with root-to-tip stall.**

stall zone. A typical curve of performance in terms of pressure coefficient against flow coefficient is shown in Fig. 2, together with oscillograms of mass-flow rate against time for the tip and root sections. There is little radial variation of stall pattern, and from the pressure-coefficient curve it can be seen that the instigation of stall is accompanied by a sharp drop in pressure ratio, as shown at point *A*, as compared with the gradual decrease in pressure ratio for the progressive-type stall. The drop in pressure rise is of the order of one-third of the stage-pressure rise prior to stall. Further reductions in flow result in an increase in the circumferential extent of the stall zone and a gradual decrease in pressure coefficient. The stall zone may cover one-third or more of the annulus (in circumferential extent) and has an absolute rotational speed of 30 to 40 per cent of the rotor speed. A root-to-tip stall zone having been established, the flow may be considered to be divided in two parts, a low-flow or stalled part and a high flow or unstalled part, respectively. Changes in compressor weight flow may then result in changes in area or weight-flow rate of the two zones. In order to unstall this type stage, the stalled zone must be completely eliminated. Because of the freedom of change of area or of flow rate in the stalled zones, these zones will persist when the compressor weight flow is increased above the value for instigation of stall. Unstalling of the stage shown in Fig. 2 is obtained at the point indicated as *B*, and a hysteresis effect results. Thus, the stage performance curve has discontinuities between the branches.

The value of the pressure coefficient on the stalled branch of the curve in Fig. 2 was determined by conventional steady-state instruments and represents some average value of the pressure coefficient for the stalled and unstalled portions of the annulus. Because of the response characteristics of the instrumentation used, the pressure coefficient in the stall region does not necessarily represent a true average of the stage performance. The existence of this double-branch type of stage characteristic is frequently unsuspected or ignored in the presentation of single-stage performance. Discontinuities and double-valued pressure coefficients, however, must be considered in computations of multi-stage compressor performance from single-stage data and are extremely important with regard to studies of compressor stall and surge, inasmuch as a double-valued single-stage characteristic must effect a double-valued multi-stage characteristic.

General Discussion of Rotating Stall: In general, low hub-tip ratio stages or stages which approach critical conditions much more rapidly at one end of the blade span than at the other exhibit progressive-type stall. High hub-tip ratio or exit stages in which angle of attack variations along the blade span are small exhibit root-to-tip stall. Available data on intermediate hub-tip ratio or typical middle stages of a multi-stage compressor exhibit first a progressive stall which, with further reductions in flow, changes to a root-to-tip stall with the associated sharp drop in pressure and the hysteresis characteristic. An important difference between progressive and root-to-tip rotating stall is the gradual decrease in pressure coefficient associated with progressive stall, as compared with the discontinuity of pressure coefficient associated with root-to-tip stall. Of secondary importance is the fact that the frequencies of flow variation are generally much higher for progressive than for root-to-tip stall.

The rotational velocities of the stall zones are approximately proportional to the compressor speed, and in the useful speed range the multiple-zone progressive stall may result in flow-rate variations with frequencies relative to the rotor or stator blading, these being of the same order as the resonant bending frequencies of typical inlet-stage blading. Inasmuch as frequent blade failures have resulted from brief periods of operation in stall at intermediate compressor speeds, indications are that stall is a possible source of resonant excitation that may lead to blade failure.

If should be noted that the rotating-stall pattern is not always as sharply defined as indicated in the stages discussed, and at least one stage has been tested in which no rotating stall existed. This particular stage has an extremely high camber at the blade tip. Thus, although rotating stall appears to be prevalent, it is not necessarily the only type of stall which can exist.

STALL OF MULTI-STAGE COMPRESSORS

The second phase of this study of axial-flow-compressor stall is consideration of the effects of the single-stage stall characteristics on the multi-stage stall problem. For operation of high-pressure-ratio multi-stage compressors at 50 per cent of design speed, flow restrictions in the latter stages limit the flow, and result in stalling of the front stages. Despite front-stage stalling, however, the compressors do have an appreciable flow range of usable operation at low speeds. It is evident, therefore, that stage stall is a necessary but not a sufficient condition for multi-stage compressor stall.

Computed Multi-stage Performance: The results of preliminary experimental studies suggest an analytical study of overall multi-stage compressor stall and off-design operation based on a knowledge of individual stage characteristics. An approximate knowledge of variations in operating conditions of each stage can be obtained over the complete range of multi-stage operation by a stage-stacking technique. A representative analysis was made, therefore, for a hypothetical 12-stage compressor having an overall pressure ratio of 7·75 at the reference or match point. For this analysis, stage performance was generalised in the form of pressure coefficient and adiabatic efficiency against a flow coefficient based on mean radius velocities, and the overall compressor performance was computed with a stage-by-stage technique. Single-stage data suggest that high-flow, high-pressure-ratio inlet stages will probably have progressive stall; on the other hand, middle stages will have progressive stall followed by root-to-tip stall, and exit stages will have only root-to-tip stall. The first four stages, therefore, were assumed to have a progressive stall, and the stage characteristics shown in Fig. 3 were chosen. For stages 5 through 8, an initial progressive stall followed by a root-to-tip stall at a lower flow coefficient was assumed; this assumption gave the stage characteristics indicated. The last four stages were assumed to have only a root-to-tip stall, and thus the selection of the rear-stage-performance characteristics was as shown.

The curves for all stages were identical except for the stall characteristics. A reference speed was chosen and the area ratio through the compressor determined so that all stages operated at a flow coefficient of 0·69 at the reference condition. Overall compressor performance was then computed for speeds from 50 to 100

per cent of the reference speed in 10 per cent intervals. It should be noted that the stacking technique based on generalised stage curves does not consider decreases in either the stage range or efficiency at extremely high Mach numbers, nor does it account for changes in performance of a given stage as a result of non-steady or non-uniform flow imposed by adjacent stages. This technique is used primarily to determine the relative matching of the stages for off-design performance points for the multi-stage compressor and, in consequence, to determine how peculiarities in stage characteristics resulting from instigation of rotating stall may affect overall performance.

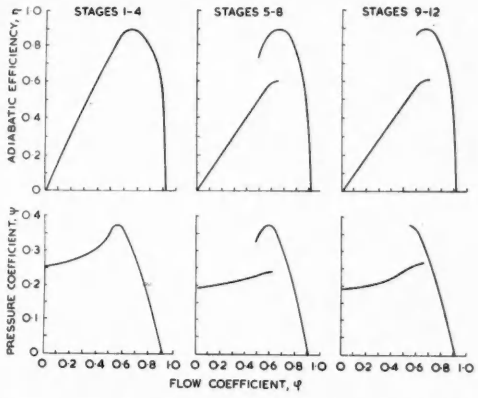


Fig. 3. **Stage performance parameters for stacking analysis.**

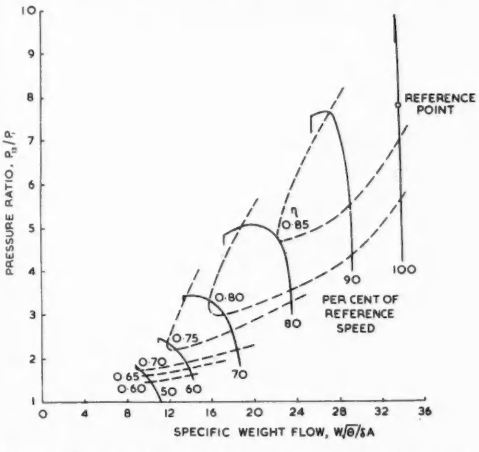


Fig. 4. **Computed multi-stage compressor performance.**

The computed compressor map in terms of pressure ratio against specific weight flow (flow rate per unit of annulus area) is shown in Fig. 4. An appreciable difference in peak efficiency at high and low speeds is indicated by the contours of constant efficiency. Efficiencies of slightly over 0·70 were computed for 50 per cent speed, whereas a peak efficiency of over 0·85 was obtained for the reference speed. The islands of high efficiency at the reference speed are broader than those generally measured experimentally; this difference can be attributed to the fact that neither the range of high individual stage efficiency nor the maximum of stage efficiency was reduced as Mach number was increased.

Stage-wise Variation of Flow Coefficient: One of the

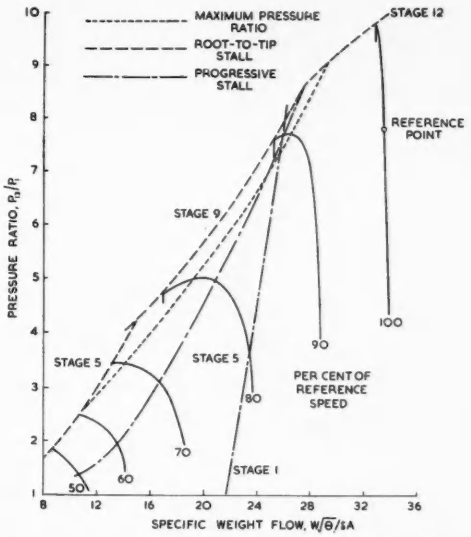


Fig. 5. **Stage stall limits for multi-stage compressor.**

principal results of the stage-stacking analysis is the determination of the particular stage responsible for the discontinuity in overall pressure ratio at each speed.

In order to study the speed ranges over which various stages are stalled, the overall performance map has been replotted (Fig. 5), and lines of first-, fifth-, ninth-, and twelfth-stage stall have been added. The discontinuities in overall pressure ratio for speeds of 50, 60, and 70 per cent of the reference value are the result of instigation of root-to-tip stall in the fifth stage. For speeds of 80 and 90 per cent, the discontinuity is caused by the ninth stage. At higher speeds, root-to-tip stall of the twelfth stage causes the discontinuity in the overall pressure-ratio curve. The magnitude of pressure drop shown is the result of stall of only one stage. As a result of flow fluctuations resulting from root-to-tip stall, however, adjacent stages close to stall would probably be adversely affected, and the magnitude of pressure discontinuity would be much greater than that indicated.

All points to the left of the inlet-stage stall-limit line indicate the existence of a progressive stall in one or more of the inlet stages. All the flow range for 50, 60, and 70 per cent speeds and most of the flow range for 80 per cent speed are in the range of inlet-stage stall. The fifth-stage progressive-stall limit line shows that progressive stall exists at least back to the fifth stage over the low-flow range for speed up to 90 per cent of reference. As shown in Fig. 5, therefore, progressive stall exists in many of the inlet stages at low speeds for all flows, and it also exists for low flows at intermediate speeds. Discontinuities in overall pressure are instigated by root-to-tip stall of the middle stages at low speed and the latter stages at high speed.

General Considerations: Effects of progressive-type stall on altering the effective performance curves of succeeding stages have not been considered in this analysis. Such interactions would probably result in some deterioration of stage performance, but would not appreciably alter the general trends of the pressure ratio against weight flow at a constant speed. At intermediate speeds, where a large number of stages is approaching stall simultaneously as flow is decreased, root-to-tip of middle stages may occur at a higher value of flow coefficient than normally, as a result of the progressive stall of early stages. This situation would cause the discontinuity of overall pressure ratio to

occur at a higher value of specific weight flow and at a lower pressure ratio than indicated by this analysis.

With regard to the relations to compressor surge of the sharp discontinuities of pressure ratio shown in Fig. 5, it may be said that, whether or not the compressor surges, these points represent a lower limit to the useful flow range at any speed. It will be noted that, at speeds of 80 and 90 per cent, the discontinuities in pressure ratio occur at values of pressure ratio less than the peak. On the basis of stability criteria, surge may be encountered at approximately the flow for peak pressure ratio; therefore, a curve through the peak pressure-ratio points at various speeds has been included in Fig. 5. The curves of neither overall stall point nor peak pressure ratio indicate the kinks in the compressor stall or surge line which are prevalent for high-pressure-ratio axial-flow compressors. Observation of regions where progressive-type rotating stall prevails, however, suggests that stage interactions may induce root-to-tip stall in inlet stages, which might readily cause kinks in the compressor stall-limit line. Variations in stage matching and stage characteristics may also be responsible for the undesirable kinks. As indicated in this analysis, a large group of stages in the middle of the compressor is operating near their individual stall points at intermediate speeds, and, if any one stage stalls, a serious deterioration of many stages may be precipitated. Further studies of stage interactions are necessary to predict completely compressor stall characteristics at low and intermediate speeds. A stall may cause discontinuities in individual stage characteristics and result in sharp discontinuities in the overall compressor performance curve. A definite limit of useful operating range of the compressor at any speed, regardless of whether surge occurs, is thus directly attributable to the discontinuity in the performance of one or more of the individual stages.

(To be concluded)

U. S. A.

A Method of Measuring Case Depth

By D. J. WRIGHT. (From *Metal Progress*, Vol. 64, No. 6, December 1953, pp. 86-88, 3 illustrations.)

NUMEROUS methods are employed throughout industry to measure the depth of carburised cases on steel, largely because there is apparently no standardised procedure which is acceptable to all. Articles have been published describing various "approved" or "standard" procedures, but generally each company or industry has preferred to apply its own particular technique, based on its own interpretation of case depth and the availability of either shop or laboratory equipment.

Until approximately seven years ago, the heat treating division of Caterpillar Tractor Co. encountered inconsistencies, not always explainable, in case depths on test bars. Each such instance involved considerable extra expense because of additional testing and destruction of piece parts. An analysis of the problem revealed that very often the fault was in the method of determining case depth rather than in the carburising operation. At that time, therefore, it was decided to define the most applicable and economical method of checking this metallurgical quality and standardising accordingly. The objectives which would have to be met were as follows :—

(1) Determinations must be accurate and repetitive and the method must eliminate differences of judgment between persons making the determination.
(2) Determinations must be made quickly, so that "in process" heats could be checked at any time without delay.
(3) The procedure must be simple, so as not to demand highly skilled technicians, a requirement particularly important during week-end or holiday operation.
(4) Equipment should be relatively inexpensive and durable, and the cost of individual determinations should be as low as possible.
(5) The method used should represent a measure of the control of the carburising process, and should not attempt to evaluate other variables, such as the material or design of the piece.
(6) The determination should indicate a case depth which would be acceptable under a broad interpretation of the term "effective" depth of case.

Several approaches were tried during the investigation for a method which would meet these requirements. Thus, test bars of carbon and alloy steels were etched or heat-tinted to resolve the carburised case for its measurement with a special scale. Brinell or equivalent low-power magnifications, in conjunction with appropriately calibrated lineal divisions, were used to measure the case on fractured or polished and etched transverse sections. Two other ideas studied were hardness tests on step-ground and tapered bars and transverse sections, and the microscopic examination of carburised, and of carburised and hardened, bars which had been tempered at 1000° F in cast iron borings.

In keeping with the objectives stated, a combination of the first two methods, regulated so as to control the known variables and made to conform easily with all requirements, was selected. During many years of use, this method has given satisfactory results, and it is considered certain that any troubles which may have been encountered are not due to the method of checking case depth.

The procedure employed, which is very simple to apply after a brief initial instruction, is as follows :—

Test Bars:—Test bars are procured from a mill heat of known characteristics and in sufficient quantity to last a number of years. The cold finished bars are $\frac{7}{16}$ in. square and $3\frac{1}{4}$ in. long, and a hole is drilled near one end on those requiring suspension in the furnace. Chemical composition of the bars is :—0·17 to 0·20 C, 0·80 to 0·90 Mn, 0·17 to 0·22 Si, 0·040 S, 0·020 P, 0·12

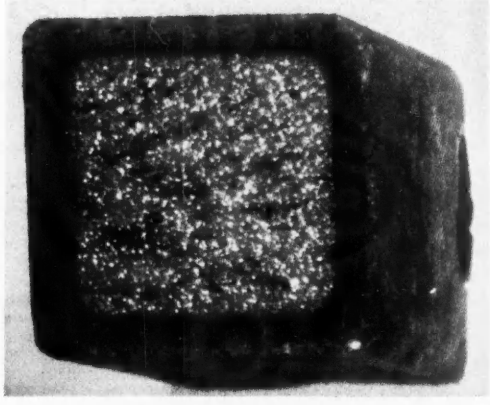


Fig. 1. **End view of test bar after being fractured and etched (×6 approx.).**

Ni, 0·03 Cr, 0·02 Mo, 0·05 Cu, 0·15 total Ni plus Cu ; all but the first three are maximum values.

The A.S.T.M. grain size is No. 1 to 4, with a maximum spread of two numbers. The test bars are run with the production parts in every furnace charge and at regular frequent intervals on continuous furnaces. Test bars which have been checked for case depth in the manner to be described are used to accept or reject production material with which they have been carburised. Use of test bars of uniform carburising characteristics is necessary, in order to isolate the variable being checked to that of the carburising routine itself.

After cooling from carburising to at least a " black " heat, bars are loaded into a small electric furnace equipped with a circulating fan. Up to 20 pieces are loaded at one time in an alloy pan. This furnace is used for this work only and is maintained at 1440° F ± 5° F at all times. Heating time is held to 22 min ± 30 sec, and no protective atmosphere is used. Test bars are then quenched until cold in a full flow of water at 75° F ± 5° F.

Fracturing and Etching:—The bars are broken transversely at approximately their mid-point and etched in 10% nital for 8 sec. Care must be exercised to use fresh nital which has not been exposed to air for more than 2 hr. The etched fracture is rinsed in water and dried with compressed air, resulting in a clear, definite line of demarcation between case and core.

Reading Case Depth:—The instrument developed for determining depth is of a simple, inexpensive design, yet it has eliminated the difference between individuals making the determination and the natural inclination of people to " interpret " the case measurement as they may happen to see fit. The principal parts of this instrument are an illuminated Bausch & Lomb Brinell microscope and a dial indicator mounted on a metal base. When the test bar is placed in the instrument, the hairline of the microscope is " zero-ed " to its edge. The bar is moved so that the hairline coincides with the first free ferrite. The case depth is then read directly on the dial indicator.

From the investigational work effected, it was found that a reading to the first grain of free ferrite would coincide closely with a 0·40% carbon content and a hardness of Rockwell C-50. On this instrument,

(*Concluded on page* 68)

GERMANY

Improved Absorption Silencing for Ventilating and Exhaust Systems

By O. GERBER. (From *Konstruktion*, Vol. 5, No. 11, November 1953, pp. 363-367, 15 illustrations.)

This article indicates various considerations which have led to improved designs of absorption-type silencing equipment, by means of which a noise reduction of up to 380 db/metre is obtainable at low frequencies. The applications described include silencers for ventilating ducts, fans, and exhaust systems of internal combustion engines.

THE importance of noise reduction is well recognised and silencing arrangements are now found in such widely different applications as buildings, compressor units, aircraft, stationary and vehicle engines, etc. An effective means of dealing with noise problems is provided by absorption methods. These allow the sonic field to expand into absorbent materials, in which the vibrational energy is dissipated in the form of heat.

SOME APPLICATIONS OF ABSORPTION SILENCING

In architectural acoustics (for office buildings, concert halls, theatres, schoolrooms, etc.), a reduction in noise level is obtained by providing the walls with layers of porous materials in order to alter the reverberation period, i.e., the time required for noise to decay to a very low level. The reverberation is measured as an attenuation per unit time, in decibels per second.

In ducts and pipe systems, absorption silencing can be obtained with straight-through arrangements, which reduce the noise as it propagates along a pipe. This "longitudinal attenuation" can be determined in decibels per metre of pipe length.

The treatment of ducts for noise reduction is usually as follows:—The duct cross-section being left unchanged, part of the periphery of sound-reflecting wall material is provided with perforations leading into an outer space filled with absorbents, such as glass-, steel-, or aluminium-wool. Alternatively, a reduction in cross-section is permissible, within certain limits, and the duct is lined with sound-absorbent layers of these materials.

The thickness of the porous layers does not usually exceed 10 cm. The longitudinal attenuation N_L, i.e., the noise reduction per metre of pipe length, can be estimated approximately by means of Piening's formula :—

$$N_L = 1{\cdot}5\ \alpha\ P/S \quad [\text{db/m}]$$

where α = absorption coefficient of material, P = sound-absorbing perimeter of duct [m], and S = free area of cross-section [m²]. Since the value of α is known, the above formula gives the specific attenuation, and from this the length of the muffling section is determined for the amount of noise reduction desired.

For the ducts of ventilating, evacuating, and air conditioning systems, the above procedure has usually been satisfactory, as in these applications pipe lengths of up to 5 metres are generally available for silencing, and the noise level of medium or large-size installations in most cases does not exceed 90 phons. Considerably different conditions occur, however, when dealing with powerful high-speed turbo-chargers tested in aircraft engine factories. Moreover, engine exhaust systems have also been a difficult problem. In these applications, the absorption silencers hitherto developed have not been completely successful, so that preference has been given to silencers of the reflection or interference type. Finally, the problem of noise reduction in the low-frequency range has never been satisfactorily solved, although it is important, since for machines such as fans (for which the fundamental frequency is given by the product of blade number and running speed) and internal combustion engines (fundamental frequency determined by number of cylinders, strokes per cycle, and engine speed), the principal ranges of excitation are always situated in the lower tones.

IMPROVED ABSORPTION SILENCING METHOD

It was necessary, first, to determine whether the absorption-silencing procedures so far used had reached maximum efficiencies, and if not, to investigate means for further improvement of the noise level and frequency characteristics of longitudinal attenuating systems. Theoretical work to this effect was carried out by Cremer, and this was followed by an experimental investigation by the author of this report.

Before describing the results obtained from this research, it is useful to consider some of the factors which determine the damping of sound in ducts.

In a duct with rigid walls, it is evident that a sound wave can propagate without loss of energy. If the duct, however, is designed with flexible inner surfaces, these will be subjected to the vibratory velocity $\overline{v}$ of the air or gas particles impinging on the walls with a sonic pressure $\overline{p}$. The ratio $\overline{p}/\overline{v}$ is known as the "acoustic impedance" $\overline{Z}$ of the wall. The bar used on these symbols denotes that they are complex numbers, i.e., vibration vectors. Thus, the acoustic impedance $\overline{Z}$ consists of a resistive component R and a reactive component X, and by means of these it is expressed as $\overline{Z} = R + jX$. Its reciprocal is termed the "acoustic admittance" of the wall: $1/\overline{Z} = A + jB = \overline{v}/\overline{p}$.

As sonic pressure acts on the walls, these will vibrate with a velocity $\overline{v}$ determined by their impedance. If the walls are made sound-absorbent, some of the energy will be dissipated in the absorbent material, and this will therefore reduce the total quantity of sonic energy passing through the length of the duct, so that "longitudinal damping" is obtained.

The longitudinal attenuation N_L can be calculated as follows:—The sonic power (i.e., the energy per second) W passing through a rigid-walled duct is

$$W = \overline{p}^2 S/(\rho c),$$

where $\overline{p}$ = sonic pressure, S = area of duct cross-section, ρ = mass-density of gaseous medium, and c = velocity of propagation of sound in this medium. Therefore,

$$\overline{p}^2 = (W/S)\,\rho c \qquad .. \qquad .. \qquad (1)$$

In a sound-absorbing duct (assumed to be of rectangular cross-section, with a perimeter $P = 2(h+b)$, where h = height, and b = width of duct), the real component of the power dissipated in the walls over a length dx of the duct is

$$-dW = P\,\overline{pv}\,dx = P\,A\,\overline{p}^2\,dx \qquad .. \qquad (2)$$

since $\overline{v} = A\overline{p}$, where A is the real component of the wall admittance $Y = A + jB$. Substitution of the expression for $\overline{p}^2$ in eq. (2) gives

$$-dW/W = P\,A\,\rho\,c\,dx/S$$

and, as $\log_e (W_0/W) = [\log_{10} (W_0/W)]/0{\cdot}435$, the decibel attenuation N_L per unit length of the duct is finally obtained as

$$N_L = 10 \log_{10} (W_0/W) = 4{\cdot}35\,P\,A\,\rho\,c/S \qquad (3)$$

Eq. (3) shows that the noise reduction N_L depends on the ratio P/S, as already indicated by Piening's formula. In addition, it shows that N_L depends on the nature of the wall impedance, even though, for simplicity, only the real component of the admittance has been considered.

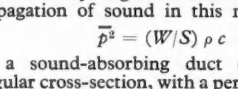
An important conclusion derived from eq. (3) is that the circular cross-section is the most unfavourable shape for noise reduction, as, for a specified value of S, it gives the smallest P/S ratio. The most favourable shape is that of a narrow rectangular cross-section. In this way, moreover, only the two large sides need to be made sound-absorbent. As $P = 2b + 2h$, and $S = bh$, it can be seen that, where b is much greater than h, the P/S ratio is inversely proportional to h/2:—

$$P/S = (2/h) + (2/b) \simeq 2/h$$

Consequently, h should be chosen as small as possible, to obtain high attenuation values corresponding to short silencer lengths.

A second important conclusion concerns the nature of the wall impedance. Eq. (3) indicates that the wall impedance should tend towards zero, so as to attain infinitely high attenuation values. However, this very simple conclusion cannot be fulfilled in practice, since it is not possible to develop a wall impedance based on vacuum. Moreover, the results expressed in eq. (3) are necessarily based on simplifying assumptions which disregard some of the factors occurring in this problem. Consequently, there remained no other choice but to try to determine the finite, attainable value of wall impedance giving maximum attenuation.

It was found possible to determine for all frequencies the complex value and composition of wall impedances resulting in maximum attenuation. It was also estab-



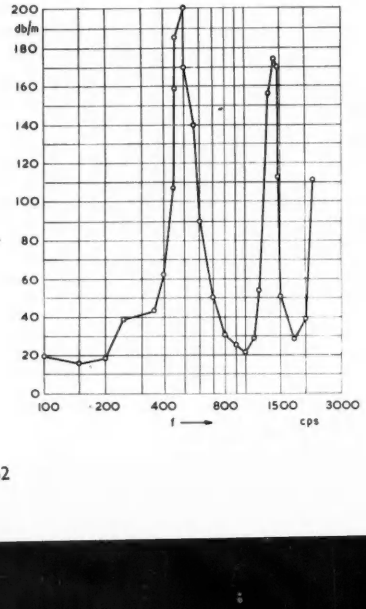
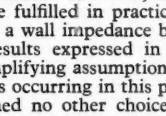


Fig. 1 (left). **Schematic arrangement of silencer with wall impedances for maximum attenuation at 460 cps, and corresponding frequency characteristic.**



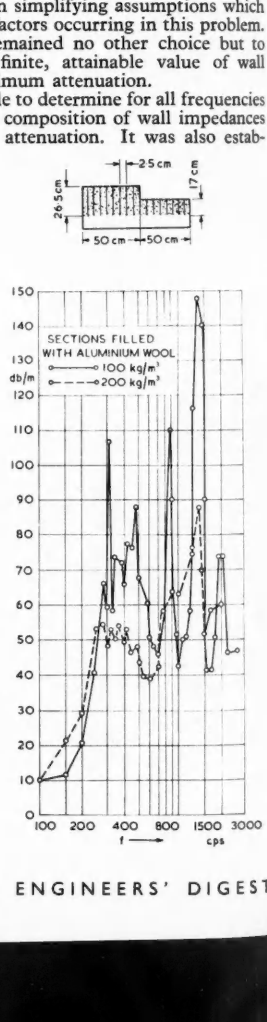


Fig. 2 (right). **Silencer with wall impedances for two frequencies.**

lished that the wall impedance should always include an "elastance" component; furthermore, both the real and the imaginary component should increase linearly with frequency.

If it were possible to make a composite wall construction with the correct impedance for all frequencies, then a duct of 10 cm in height, incorporating this impedance in its large-side surfaces, would reach theoretical attenuation values which seem improbably high, *viz.*: 380 db/m at low frequencies, and 300 db/m at 1000 cps, with an appreciable falling-off occurring only towards the high frequencies, when the duct width becomes large relative to the wavelength of the sound parameters, so that the sound has an increased possibility of passing through the ducts without touching the absorbing walls.

As there was no conception at first of the technical design features of acoustical impedances which would reach such remarkable values, experimental work was undertaken to investigate the possibility of producing silencing arrangements in the form of wide passages 10 cm high and 1 m long, which would be effective, in the first applications, for certain frequencies and their harmonics. The tests were successful and the attenuations obtained attained 200 db/m.

On the basis of the theoretical and experimental work carried out, it is now possible to indicate how a wall impedance should be designed, in order to eliminate a specified frequency:— (1) The imaginary component of the impedance should be not quite as low as the dimension of a quarter wavelength of the frequency to be attenuated. (2) The walls should include porous sound-absorbent materials, as loosely packed as possible, so as to obtain the required value for the real component of impedance at the specified frequency. (3) The arrangement should be designed to prevent propagation of sound in the wall materials parallel to the length of the duct, i.e., it should include transverse wall-sections located as closely together as possible. These built-in wall-sections may consist, for instance, of sheet metal or hard fibre plates, which reflect sonic energy into the absorbent material.

A silencing duct of this type, used in the first tests for the attenuation of sound at a frequency of 460 cps, is shown in Fig. 1, which also gives its frequency characteristic.

It is evident that this silencer would not be suitable for practical purposes, despite its high attenuation at certain frequencies, owing to the very low values of attenuation occurring between the peaks. Therefore, another silencer was built, which was designed to cope with two specified frequencies and their harmonics. This arrangement consists of two ducts, with a maximum length of 50 cm for each frequency. The second element was designed to give maximum attenuation in the gap in the frequency characteristic of the first element. The arrangement thus developed, together with its frequency characteristic, is shown in Fig. 2.

Although the noise-reduction characteristic has many sharp peaks and irregularities, it shows, however, a mean attenuation of about 50 db/m over a wide frequency range. Additional effectiveness towards the low frequencies was obtained by inserting a low-tuned resonator between the two duct elements. The resulting arrangement (see Fig. 3) gives a mean attenuation of 60 db/m over the range from 200 to 2000 cps. This wide-band attenuation has probably never been obtained previously with 10-cm ducts.

From the above, it is apparent that the improved method of absorption silencing offers extensive possibilities in practical engineering applications.

EXAMPLES OF APPLICATIONS

Noise reduction in ducts of ventilating and pumping systems: In most cases, the ducts are of ample dimensions, so that there is a possibility of reducing even

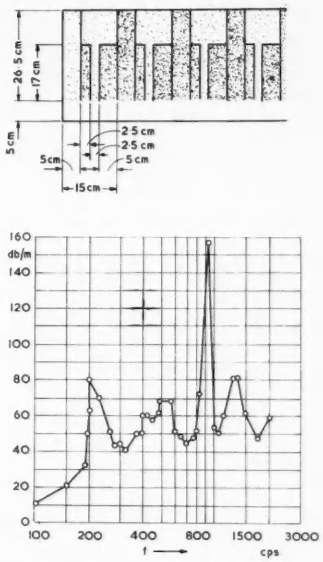


Fig. 3. **Silencer arrangement for wide-band attenuation, and corresponding frequency characteristic.**

very low frequencies. However, a difficulty occurs as regards proposed restriction of duct height, in order to obtain narrow rectangular passages with large b/h ratios. Generally, the designers of flow-type machines will not allow flow velocities to exceed 6-7 m/sec, so that a duct height of 20 cm (instead of 10 cm) has to be used. It is then necessary to use 2 metres of duct length (instead of 1 m) to obtaiin a 60-db attenuation. This 20-cm height also has the disadvantage that it does not permit high acoustic frequencies to be damped to the desired extent, and these are soon noticeable in the frequency spectrum.

Preparations for duct noise reduction proceed as follows:—Data are received from the designers regarding duct cross-sections, volumetric rate of flow, and maximum permissible air flow velocity. Measurements are obtained of the frequency spectrum of the noise to be reduced; alternatively, the principal frequency is estimated by means of the relation

$$f = z\,N/60 \quad [\text{cps}]$$

where z = number of blades of ventilating fan and N = running speed [rpm], so as to determine the pitch of the noise, i.e., the range of frequencies to be considered. The noise level is measured or estimated from previous experience.

The first modification is then made to the duct cross-section, which is changed from an approximately square shape to a cross-section consisting of one or more narrow, rectangular passages suitable for silencing purposes, within the required limit for flow velocity. These parallel passages or channels are in most cases so designed as to have their maximum dimension in the vertical direction, and separated by walls with the required impedances on either side of each duct. The walls may consist of several sections of different impedances, so as to reduce a number of frequencies. Usually, a silencer length of about 1·5 m is sufficient to achieve a reduction of 40 db, with channels 20 cm high and a b/h ratio between 5 : 1 and 10 : 1. This attenuation is usually adequate, as the long duct system also contributes somewhat to noise reduction. A silencer, 1 m long, 2 m wide, and provided with two channels each 25 cm high and approximately 1·8 m wide, was built on these principles for a ventilating system with a

capacity of 14,000 m³/hr, and gave an attenuation of 25 db at 100 cps, rising to a maximum of 52 db at 600 cps, and then dropping to 30 db at 3000 cps, and 10 db at 6500 cps, the average attenuation being 40 db over the entire range. Although the theoretical value was 60 db, this overall reduction of 40 db is very satisfatory, in view of the difficulties in obtaining the appropriate absorbent materials for loosely-packed sections.

Silencers of this new type have also been designed for reducing the loud noise from the intake and delivery of a large compressor set operating with air flow velocities up to 100 m/sec. Unfortunately, the test results are not yet available.

Exhaust noise silencing: The main limitations in this application occur with road vehicles, and concern the size, weight, and cost of the absorption silencer. Furthermore, in all exhaust systems, there is the problem of maintenance, so as to avoid the accumulation of soot and other deposits.

It is certainly possible, however, to find satisfactory solutions for these problems. On exhaust systems, the absorption type silencer gives very good attenuation values, with little or no effect on engine performance.

Fairly favourable conditions as regards silencing arrangements are available in heavy lorries and motorships, where space and weight are not primary considerations. These allow absorption silencers giving highly effective results to be provided.

The exhaust silencer for heavy road vehicles is similar in its general construction to those developed for ventilating plants. The circular exhaust pipe is converted into a narrow, rectangular channel and the wall impedances are suitably designed for the range of frequencies to be covered. An " air cushion," serving as a low-frequency filler, is included to cope with the lowest frequencies.

Exhaust systems, as compared with ventilating ducts, have the great advantage of having much smaller cross-sections, so that attenuation can be obtained with fairly short silencer lengths. There is a disadvantage, however, in connection with the hot exhaust gases, which result in higher velocities of sound propagation and hence increase the wavelengths of the audible noise.

Fig. 4 shows an absorption silencer for a six-cylinder diesel engine of about 100 hp used in a river patrol boat. This silencer is also suitable for overland vehicles.

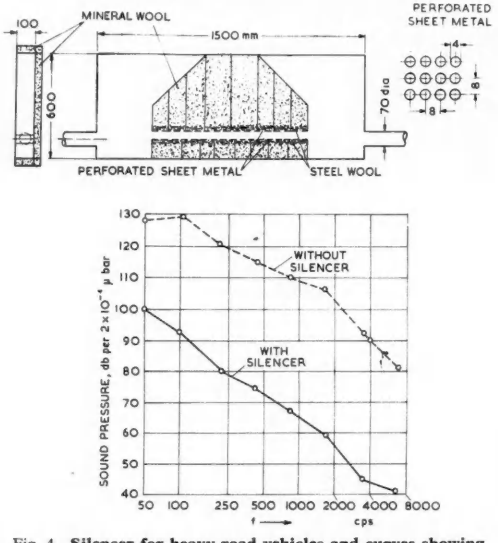


Fig. 4. **Silencer for heavy road vehicles and curves showing attenuation obtained.**

With a silencing length of 90 cm and a channel 5 cm high, this silencer gives the very good wide-band attenuation of 40 db, reducing the noise level from 100 phons to 70 phons.

Comparing again practical results with theory, it can be seen that much less satisfactory agreement is obtained than in the case of ventilating ducts. A channel with a height of 10 cm would be expected to give a noise reduction of 60 db, according to theory and previous experiments. The problem with which the acoustics engineer is now confronted is to determine the reasons for this difference. The packing density of the porous materials is, of course, one of the factors to be considered. However, the major factors, in all probability, are the complex non-linear features of the sonic field of the pulsating gases, which were not taken into account in the theoretical investigation. The laboratory tests were made using pure sinusoidal tones in still air. There is not yet, unfortunately, sufficient information on the sonic field of exhaust gases of internal combustion engines, so that, for the present, a reduced value of 40 db for the average attenuation is the best that can be attained.

An absorption silencer has also been developed for motor-cycles. The unit is of rectangular shape (length 50 cm, width 4 cm, and height 30 cm), with inlet and outlet pipes at its lower ends. It can readily be installed in this form in motor-cycles with faired-in structures, as well as on scooters and other two-or three-wheeled vehicles. For sports machines, the question is where to instal it, and whether its shape should be modified or not. These problems will certainly be solved when an absorption silencer specially designed for exhaust noise is developed.

The exhaust silencer for motor-cycles was tested on a single-cylinder two-stroke engine, and was found to give a maximum attenuation of 27 db at 230 cps, with minimum values of 22 db for frequencies on either side of this peak. The characteristic is therefore fairly flat. Again, this silencer gives a noise reduction equal to approximately half the theoretically expected value only. However, it reduces the exhaust noise nearly to the level of the intake. With a further reduction of 5 db, engine noise will be the predominant factor. It can safely be assumed that such a reduction will be obtained, as this was the first experimental silencer built for this application, without appreciable increase in size.

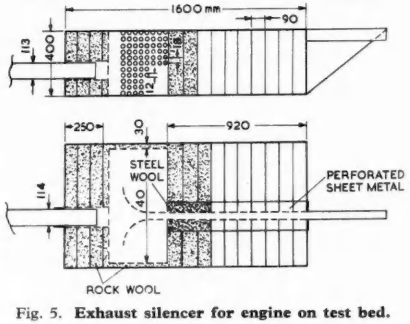


Fig. 5. **Exhaust silencer for engine on test bed.**

Finally, Fig. 5 shows the inside arrangement of an exhaust silencer designed for stationary sets, test engines, etc. As there is no special restriction on size, the silencer can be designed to give any required attenuation, for all frequencies, without affecting engine performance. Silencers of this type can always fulfil theoretical specifications by being provided, for instance, with a large number of closely spaced transverse wall-sections. The cost in this case is of secondary importance, as the same silencer may be used to test a number of engines.

New Czech Boiler Plant

By R. DOLEZAL. (From *Strojirenstvi*, Vol. 2, No. 11, 1952, pp. 487-491, 3 illustrations.)

New methods of steam regulation, of utilisation of the heat content of the slag, and of preheating combustion air are described in this article.

THE boiler installation described in this article was designed to supply up to 220 tons per hour of steam at a temperature of 1004° F and a pressure of 1630 psi, one requirement being that it should operate, in conjunction with a 50-MW steam turbine, on any type of coal from Poland or from the Ostrava region of Czechoslovakia.

In the design and construction of this boiler installation, a longitudinal cross-section of which is shown in Fig. 1, several new ideas have been employed, these representing the results of boiler research carried out by the author.

FEEDWATER SUPPLY

The boiler is provided with two independent 3-impulse feedwater regulators. Each of these regulators is located in a separate line and can be rendered inoperative without stopping the boiler.

The feedwater is first preheated to 392° F in the economiser, which is located in the second pass between the air preheaters. After preheating in the economiser, the water is further heated in high-pressure condensing feedheaters to a temperature near to the boiling point of the water at the pressure prevailing in the boiler.

The water is fed into the front washing drum, the level of which is maintained constant and higher than in the water-steam drum; the difference in level depends on the steam output at any given moment, and increases with increasing output. As a result, the space in the water-steam drum filled with steam increases automatically with increasing boiler load, ensuring efficient separation of steam moisture by precipitation.

EVAPORATION SYSTEM

The steam generator is of the radiant water-tube boiler type, with natural circulation, and embodies a pulverised coal-fired slag-tap furnace. It is provided with a front washing drum and a water-steam drum, mounted at a lower level than the washing drum. The steam-water mixture, produced in the water walls of the combustion chamber, is fed into the water-steam drum. All the generating tubes in this drum discharge above water

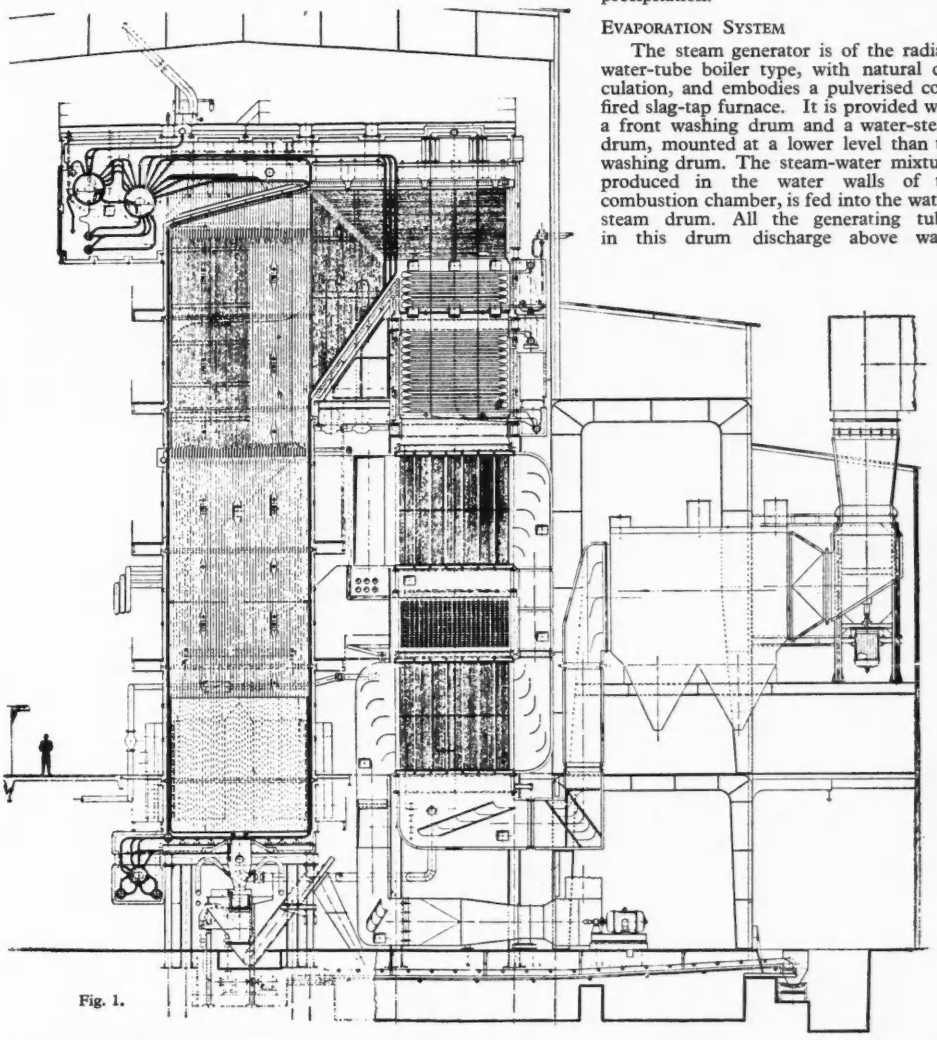


Fig. 1.

level; the water is separated from the steam by gravity. The steam separated out then passes to the front washing drum, where it is washed by the feedwater and the condensate; only the feedwater and the condensate flow through the washing drum. In accordance with Soviet practice, the steam is washed by forcing it to bubble through a layer of feedwater and/or low-concentration condensate.

From the water-steam drum the water flows through four large-diameter downcomers to the lower, smaller drum, and thence to the bottom headers of the boilers, which form the base of the tube walls and which surround the bottom of the ash melting space. Despite the fact that the furnace is of the slag-tap type, its walls consist of bare tubes, with welded-on fins. There is a continuous blow-down from the water-steam drum, where the boiler water has the highest concentration. The blow-down passes through a number of tubes provided with restrictions and special control valves. Provision is made for occasional sludge removal from the lowest point of the evaporation system, i.e., from the mud drums situated below the lower drum. The blow-down water is fed into the water circuit used for slag granulation, to make up water losses in that circuit.

For rapid, controlled starting of the boiler from cold, and for banking, the boiler is so designed that it can be heated up with steam at 285 psi and 1630 psi, drawn from other boilers.

Steam Superheating

The primary stage of superheating is carried out by radiation. From the primary superheater the steam is introduced into the steam regulation system described below.

Steam is heated to the desired final temperature in a convection-type superheater, which is made up of two sections mounted one behind the other. During its passage from the first to the second section of the superheater, the steam is thoroughly mixed in the connecting piping, so as to equalise its temperature; in addition, crossing over of the steam streams is carried out. In the first section of the superheater, combustion gases and steam flow in counter-current; in the second section, however, both flow in parallel, so as to reduce the temperature of the tube walls as nearly as possible to that of the steam temperature.

To eliminate excessive differences in temperature of the steam flowing from the individual tubes which make up the superheater, heat absorption in the individual parts of the superheater, particularly in the last section, is made as low as compatible with sufficient stability of steam flow and maximum utilisation of material.

Steam Temperature Regulation

The basic elements of temperature regulation of the superheated steam are high-pressure surface condensers.

For producing a high-pressure condensate, part of the washed, saturated steam is fed from the drum (1), as shown in Fig. 2, through pipe (6) to the condensing feedwater heater (7), in which the saturated steam is condensed at its full pressure by the colder feedwater flowing through pipe (8). The condensate produced is stored in reservoir (9) and is fed *via* pipe (10) to the control valve (11), which regulates the injection of the condensate into the steam in the injector (4). The high-pressure condensate produced from the saturated steam is of the same quality as the superheated steam; consequently, the injection of this condensate cannot cause deterioration in the quality of the superheated steam.

The condensate in reservoir (9) is at the same pressure as the saturated steam in drum (1). The condensate is injected into the steam by the injector (4) at a pressure equal to the pressure loss of the steam in the first section of the superheater, plus the static pressure of the column of condensate above the injector and the suction effect of the injector. As a result, no injection pump is required in this design.

The suction effect of the injector depends on the rate of flow of the steam at any particular time and therefore stabilises the regulation process in the event of load changes. In effect, it acts as an additional impulse, controlled by the rate of steam output.

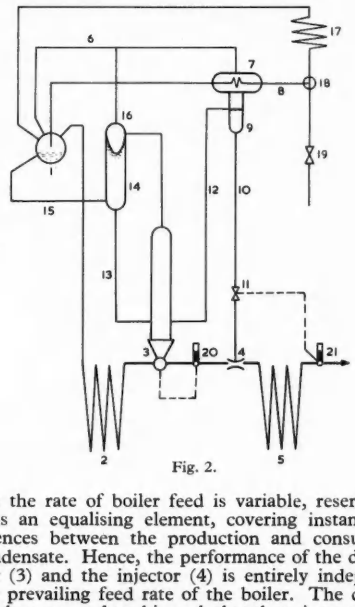


Fig. 2.

As the rate of boiler feed is variable, reservoir (9) acts as an equalising element, covering instantaneous differences between the production and consumption of condensate. Hence, the performance of the desuperheater (3) and the injector (4) is entirely independent of the prevailing feed rate of the boiler. The quantity of condensate produced is such that there is an adequate surplus after covering the consumption of the injector and the desuperheater.

The condensate which is not consumed by the injector overflows from reservoir (9) through pipe (12) into the desuperheater (3), where a portion of the condensate evaporates, owing to the heat drawn from the cooling steam, and the mixture of water and steam is conveyed by natural circulation to the cyclone separator (14).

As more condensate is fed from reservoir (9) into the desuperheater (3) than evaporates in the desuperheater, the surplus flows from separator (14) through pipe (15) into drum (1); thus, the desuperheater is continuously flushed by the condensate.

The steam from separator (14) is fed through pipe (16) back to pipe (6) and to the condensing feedwater heater (7). Thus, deposition of solids in the desuperheater from condensate flowing through, cannot take place, as, in the small steam circuit comprised by the above elements, it is always the condensate from the same steam which circulates.

Feeding of the desuperheater with a high-pressure condensate, virtually free from solids, eliminates the formation of sludge and consequent corrosion. Even in the event of a defect developing in the desuperheater, superheated steam cannot be contaminated, and there is no danger of deposition of solids in the turbine, should it be necessary to keep the boiler running in an emergency.

The boiler is fitted with a steaming economiser (17); the water flowing from the feedwater regulator (19) passes through the 3-way valve (18), whence it flows partly into the economiser and partly into the condensing feedwater heater (7). Valve (18) is set during test operation to a position ensuring a sufficiently high output of condensate by the feedwater heater, irrespective of the quality of coal used, and remains

permanently in that position during normal operation.

The equipment for steam temperature control consists only of pressure vessels and tubes. The number of moving parts is reduced to a minimum, i.e., to the 3-way regulator of the desuperheater (3) and the control valve (11).

The cooling effects of the desuperheater and injector (4) are complementary. As a result, the quantity of condensate consumed by injector (4) is reduced, and the steam temperature at the first superheater outlet is limited.

The steam temperature behind the desuperheater is controlled by thermostat (20) and is maintained constant. The temperature setting of the desuperheater is chosen sufficiently high to ensure that the design steam temperature is attained behind the second superheater stage, even in the case of reduced load. In practice, this corresponds to the steam temperature at the outlet of the primary superheater (2), at the lowest boiler output at which the rated steam temperature is still to be obtained, using a fuel giving the highest steam temperatures.

The injector (4) is controlled by thermostat (21), *via* control valve (11), and completes the cooling of the steam to such an extent that, in the secondary superheater, it is heated to the specified temperature. In the case of very low boiler loads, when the steam need not be cooled at all, the entire condensate produced by the feedwater heater flows through the desuperheater directly into drum (1).

A safety-valve is fitted behind the superheater, to protect it from damage in the case of failure of the control system; this valve opens when the steam temperature exceeds the permissible limit, in which case the flow of steam through the superheater increases by the amount released through the safety-valve, thus causing a drop in the steam temperature to the permissible limit, after which the valve closes again.

This two-stage method of superheat control has the following advantages:—Precooling of steam in the desuperheater does not cause a reduction in the quantity of steam in the primary superheater, so that excessive steam temperatures at the output end of the primary superheater are prevented; contamination of steam by water injection or by defects in the desuperheater is eliminated; the rate of feedwater supply has no influence on the performance of the steam temperature control system; stainless steel injection pumps are unnecessary; variations in injection pressure during load changes have a favourable influence on the regulation process; all water fed to the boiler passes through the feedwater heater, where it is always brought to a temperature near to the boiling point of the water in the boiler; the number of moving parts in the control system is reduced to a minimum and the thermal efficiency is in no way reduced.

Combustion Air

Combustion air is drawn from the outside or the upper part of the boiler boom by two axial fans, with variable-pitch impellers. Behind each fan there are pneumatically operated, remote-controlled dampers, which can be completely closed if only one fan is in operation. Behind these dampers, suitable connections are provided for operation by one fan, and additional, electrically controlled dampers are fitted, to ensure satisfactory air distribution to the two stages of the preheater. These electrically controlled dampers take over the main regulation, in the event of failure of the pneumatically controlled dampers.

Air is first preheated by steam in a tubular heater. As the dew points of the combustion products of Ostrava and Polish coal are below 113° F, this air is preheated to 122° F. This preheating is effected by steam produced in the evaporator from the heat released in the slag granulation process. During starting of the boiler and during severe frosts, steam at 23 psia is fed to the heater from the steam system of the power station.

After preheating to 122° F, the air is then heated by the combustion products in a two-stage all-metal tubular heater. In the first stage, the air is heated to 302-392° F. A portion of this air is extracted by two water-cooled primary fans, and serves as primary air for transporting the pulverised coal. Both these fans blow the air into a common, horseshoe-shaped duct around the boiler, from which it is drawn to the individual pulverised coal feeders.

A major portion of the remaining air is heated in the second stage of the preheater to about 752° F. Part of this air is used as a drying medium in the pulverising mills of the boiler installation, while the remaining part is fed through secondary ducts to the burners at the four corners of the combustion chamber. The distribution of air into each group of burners is regulated by remote-controlled, pneumatically operated dampers.

By feeding the combustion air into the coal mill blowers, all the air flowing into the combustion chamber passes through the intake lines preceding the fans and through the first stage of the air preheater. Consequently, the resistance of the first stage of the air preheater can be used as a reference value for the total quantity of air fed into the combustion chamber, this being particularly advantageous from the point of view of combustion process control.

In normal operation, this arrangement is beneficial in that the quantity of air fed into the boiler does not vary with starting and stopping of the pulverising mill or one of the primary blowers; the only variation in air supply is at the burners in the corners of the combustion chamber but, as combustion in such corner burners is highly turbulent, this does not affect the combustion process.

Feeding of Pulverised Coal into the Burners

The pulverised fuel is fed by a total of twelve screw-feeders, and is carried by the hot primary air at a speed sufficient to prevent clogging, even if one of the primary fans is out of action. The fuel is fed into 12 burners, located in the corners of the combustion chamber and directed towards the outflow of the slag in the centre of the base of the ash melting space. In the two rear corners there are additional burners for use with special low-grade qualities of fuel; these are located above the pulverised coal burners and are also directed towards the outflow of the slag.

In the combustion chamber, the pulverised coal is mixed with the secondary air by impact in the centre of the combustion chamber and also at the bottom of the melting space. The high speed of air flow from the burners ensures high turbulence, and consequently a very high temperature above the bottom of the slag tap furnace. Ignition of the pulverised coal during starting up is effected by oil burners with oil atomisation by steam. There is one removable oil burner located in each corner of the combustion chamber.

The main part of the combustion takes place directly above the furnace bottom. Depending on the melting point of the slag, the temperature in this zone should be between 2550 and 3100° F. On leaving the combustion chamber, the flue gas temperature should not exceed 1830° F. In the convection superheater, the flue gases cool down to about 1100° F.

After leaving the convection superheater, which is built into the second pass, the flue gas stream is divided into two halves, in which the air preheater and economiser respectively are located. The outlet temperature of the flue gases should not exceed 320° F. Electrically controlled dampers, which are required in the event of operation with one fan only, are installed in both flue gas passes. Behind these dampers the flue gas ducts are united in one line, in which pneumatically controlled dampers and electrostatic filters are provided. From

these filters, the flue gases are exhausted by two axial fans with variable-pitch impellers.

SLAG HANDLING

The molten slag flows out through an opening in the centre of the furnace bottom to a water-cooled slag trough, which maintains a minimum depth of the slag level required for protection of the piping in the furnace bottom and for thorough melting and intermixing of the slag flowing down the walls of the slag tap furnace.

Slag handling is based on the method shown in Fig. 3. The stream of molten slag is cooled rapidly by quenching sprays (1), operated with circulation water flowing from the low-pressure evaporator (2). The mixture of water and cooled slag drops into the slag tank (3) and the slag which sinks to the bottom of the slag tank is removed by a bucket conveyor to a belt conveyor.

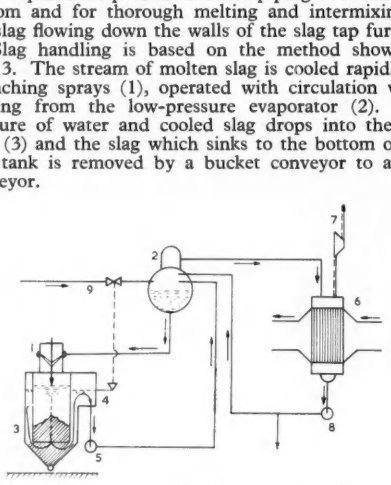


Fig. 3.

The circulation water becomes heated by its contact with the slag and is drawn off from the overflow chamber (4) by the circulation pump (5) and fed to the low-pressure evaporator (2). Owing to the low pressure, part of the circulation water evaporates ; this evaporation has a cooling effect on the remaining circulation water in the evaporator and the cooled water flows back to the quenching sprays, where the process is repeated.

The boiling of the water in the low-pressure evaporator also brings about a release of gases dissolved in the circulation water. The steam-gas mixture flows to the heat exchanger (6), where the steam is condensed by cold combustion air ; the remaining gases are drawn off from the heat exchanger by the vacuum pump (7). Depending on its quality, the condensate is either returned to the evaporator (2) by the condensate pump (8) or to the condensate reservoir.

To maintain the desired level in the overflow chamber (4), water losses must be replaced, and this is effected by blow-down make-up water at 212° F, which is fed directly into the low-pressure evaporator. The flow of this make-up water is controlled by float-valve (9). The blow-down water is hotter than the circulation water, but this does not present any disadvantages, as it becomes cooled by the heat of evaporation. The large water volume of the evaporator and of the slag tank has an equalising effect on fluctuations of short duration in the rate of slag flow.

The principal advantages of this slag-handling equipment are utilisation of the heat content of the slag for preheating combustion air, as already mentioned, and negligible consumption of cooling water, as replenishment is limited only to circulation water losses. In addition, hot, impure blow-down waters can be used for the cooling, with the advantage that, because of their high pH value, danger of corrosion of equipment is greatly reduced. These waste waters are converted into high-quality condensate by evaporation in the slag-handling circuit.

It is reported that the annual saving achieved in Czechoslovakia by the use of this slag-handling equipment is considerable.

A METHOD OF MEASURING CASE DEPTH

(*Concluded from page* 61)

accuracy of repetitive readings, between individuals as well as between different readings by the same individual, is within 0·001 in.

The end view of a test bar, after being fractured and etched, is shown in Fig. 1. Case depth is clearly defined by the darker portion, free ferrite being indicated by the light spots in the centre portion. It should be noted that, whereas Fig. 1 has a magnification of approximately $\times 6$, the actual magnification from the instrument is $\times 10$.

It is recognised that the method described here could not be applied to all shops as a means of production control, because some equipment is necessary, although it is by no means elaborate or expensive. However, it is with some hope of standardisation that this article has been prepared, since there certainly is a need in industry for closer measurement, as well as uniformity in methods employed. Perhaps variations from the foregoing would at least permit everyone to "speak the same language," as it were, and to make comparisons from the same basic beginning. If there were enough interest in standardisation, it would not be difficult to make test bars and reading instruments available, just as standard methods and equipment for hardness measurements have been available for many years.

GRAPHICAL AND NUMERICAL INVESTIGATION OF FORCED VIBRATIONS IN A NINE-MASS SYSTEM

(*Concluded from page* 54)

where a_n/a_9 (P_2) = vibration amplitude a_n/a_9 with $P = P_2$ acting on mass m_2, etc. ; $M_{[9]}$ = dynamic magnifier for mass m_9 with $P = P_1, P_2, \ldots, P_7$ (the subscript [9] indicates that all values are based on a_9 in the denominator) ; and $a_{st[9]}$ = static amplitude of m_9. Therefore,

$$a_{9(\mathrm{res})} = \sum_{(2)}^{(7)} M_{[9]} \times a_{st[9]}$$

The dynamic magnifier for the resultant vibration a_n of any mass m_n, based on a diagram beginning with a_9, is

$$M_{n(\mathrm{res})\,[a_9]} = a_{n(\mathrm{res})}/a_{st\,[9]} = \sum_{(2)}^{(7)} M_{[9]}\, a_n/a_9$$

The resonance value of any force F_n acting on mass m_n is

$$F_n = s_{st[9]} \sum_{(2)}^{(7)} M_{[9]}\, F_n/k_{89}\, a_9.$$

Moreover, using a_1 as a basis

$$M_{n(\mathrm{res})[1]} = a_{n(\mathrm{res})}/a_{st[1]} = M_{n(\mathrm{res})[9]} \times a_{st[9]}/a_{st[1]}.$$

The resultant spring force of the damper is

$$S_{89} = a_{st[9]}\, k_{89} \times \sum_{(2)}^{(7)} M_{[9]}\, s_{89}/a_9$$

and the maximum torsional stress of the crankshaft is

$$\tau_{\max} = \pm \left\{ \sum_{(2)}^{(1)} M_{[9]} \frac{\overrightarrow{a_n} - \overrightarrow{a_{n+1}}}{a_9} \right\} \frac{P}{k_{89}} \times \frac{G}{l_{\mathrm{red}}} \times \frac{r}{r_{\mathrm{red}}}$$

where l_{red} = equivalent length of shaft between a_n and a_{n+1}, P = exciting force, r = actual radius of shaft, r_{red} = reference radius, and G = modulus of torsional rigidity of shaft material.

The vector-diagram method is also readily applicable to branched systems and to systems with pendulum-type detuners.

Charts and Formulae for Evaporation Processes

By E. KIRSCHBAUM. (From *Z.VDI*, Vol. 15, No. 27, September 21, 1953, pp. 927-932, 8 illustrations.)

(Concluded from January issue)

EXPERIMENTAL CONFIRMATION OF DIAGRAMS.

The change of state of the air has frequently been represented in literature by a straight line between the air point and the surface point in the *i-x* diagram, e.g., a straight line joining the points K_L and K in Fig. 4. The error in this procedure, which assumes that the conditions are those of a mixture of two air masses in the states K_L and K, and the correctness of the relations derived in the foregoing sections of this report, have been proved by tests extended over a period of many years. The test apparatus had to be designed so as to guarantee that steady adiabatic conditions were obtained. The apparatus developed for this purpose is shown in Fig. 5.

The arrangement consists of a sheet-metal outer tube (a), containing a number of porous-stone type hollow filtering elements (b), these being hollow cylinders of 29 mm diameter and 600 mm overall length, mounted together in series. Their inner surfaces are covered with an absorbent paper (c). Temperatures are measured by means of thermocouples (d), with leads (e) inserted under the paper on the inner surface of the tubular filter unit. The gas flows through the inner section (f) of the tube, while the liquid enters the annular outer chambers (g) and penetrates radially through the wall of the filter into the inner space. The non-evaporated portion of the liquid flows out downwards through a small tube (h). The difference in the amounts of liquid entering and leaving the apparatus indicates the quantity which has evaporated over a certain length in the gas flow. By means of these indications, together with the gas temperature measurements by thermocouples, and the measurements of the state of the fresh air supplied, it is possible to determine the state of the gas at any position inside the tube. Subsequently, the apparatus described in previous reports[14,15] was improved by providing automatic temperature regulation for the liquid in the annular outer chambers, using the surface temperature indications of the thermocouples. The latter were fitted in such a way as to eliminate any effects due to heat conduction. The tests were carried out at a time when the above relations had not yet been established, so that the evaluation on this basis was made at a later date. Typical examples of the results obtained are as follows :—

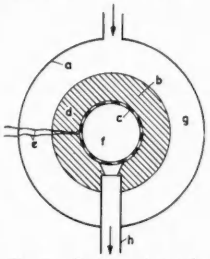


Fig. 5. **Section through tubular test apparatus.**

The experimental curves for air and water are shown in Fig. 6. With an air flow velocity of 1·6 m/sec the measurements gave an " air point " K_L in the diagram, while the thermocouples indicated a temperature of 21·3° C, corresponding to a surface point K. The points K_L and K are situated on a t^*-line of constant terminal temperature, for α/σ = 0·2 kcal/kg-°C, which corresponds with the low value for the air velocity. The measured states of the air in the tube gave test points which are situated on the solid line K_L K^*, which coincides with the τ-line through K^*. In further tests with an air-flow velocity of 6·5 m/sec (Reynolds number = 10,300), the state of the air at the inlet and outlet was determined. The corresponding points K_{Le} (inlet) and K_{La} (outlet) are plotted in Fig. 6. The other test points for intermediate states are also situated on the τ-line connecting K_{Le} to K_{La}, which intersects the saturation curve at the point K_T^*. The mean wall temperature of the tube gives the " surface point " K_m. If lines of constant terminal temperature are drawn through points K_{Le} and K_{La}, using the value α/σ = 0·23 kcal/kg-°C in view of the increased air velocity, these determine the segment of the saturation curve indicated as a thick line around K_m in Fig. 6. Theoretically, the surface point should move along this segment of the saturation curve.

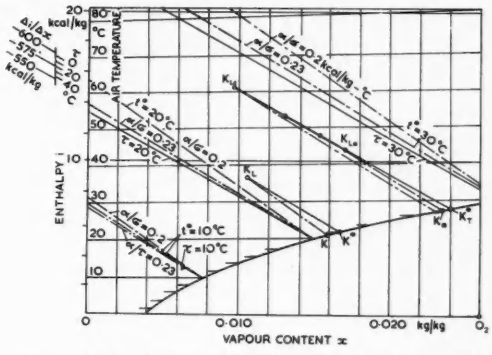


Fig. 6. **Measured states for combination of air and water.** t = air temperature ; t^* = temperature at surface of water ; τ = temperature at cooling limit ; α = heat transfer coefficient ; σ = evaporation coefficient.

The fact that the points K and K_m are different from the points K^* and K_T^*, respectively, is a proof that the change of state of the air is not given by the mixture line, but by the τ-line. The discrepancy is greater for very small or very large values of Lewis's numbers. Such cases also occur in industrial applications, as will be appreciated from the following example :

The evaporation of ethanol in air at a gas velocity of 6·5 m/sec produced the test points representing the state of the air, which are plotted from K_{Le} to K_{La} in Fig. 7. The tube surface temperatures obtained for the

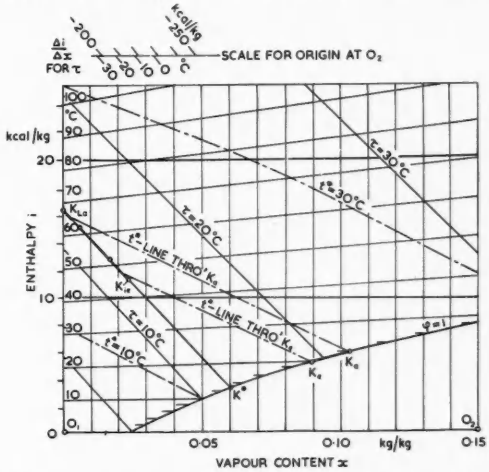


Fig. 7. **Measured states for combination of air and ethanol.**

inlet and outlet correspond to the surface points K_e and K_a, respectively. These are located at a considerable distance from K^*. Through this point, however, there is a τ-line on which the measured air points are situated. It is considerably different from a mixture line (not indicated in the figure) going from K_{L_e} to K_e, or from K_{La} to K_a. Through the two surface points K_e and K_a, however, t^*-lines (shown as chain-dotted lines) for the ratio $\alpha/\sigma = 0{\cdot}51$ kcal/kg- C, measured for this test, can be drawn. Although these t^*-lines do not pass exactly through the points K_{L_e} and K_{La}, they can be regarded as further evidence that the correlation between a surface point and an air point obtained by means of a t^*-line is correct.

Carrier and Lindsay[18] made tests in which they measured the temperature by means of a wet-bulb thermometer mounted in a dry tube. In this test arrangement the heat radiation on the wet body was an important factor, so that the assumptions regarding an adiabatic process (which are a basis of the foregoing considerations) were not fulfilled. In this case the surface temperature was higher than the terminal temperature, but it can also be lower than the latter if undercooled liquid comes into contact with hot air at the terminal temperature. In both cases, the two points K and K^* can be situated at two widely different positions, even for a combination consisting of air and water (see Fig. 8), the mixture line being invalid for the change of state of the gas.

Conditions for Obtaining Coincident t^*-Lines and τ-Lines.

The design of apparatus used for evaporation processes would be greatly simplified if the relation $Le = 1$ were generally valid. Similarly, it would be a considerable advantage if the t^*-line and the τ-line in the i-x diagram could always be made to coincide for the same temperature. As, in the foregoing, a criterion was obtained by means of Schmidt's number for determining when Le equals unity, it is also possible to set up a relation by means of which the t^*-lines and the τ-lines can be represented by a single straight line.

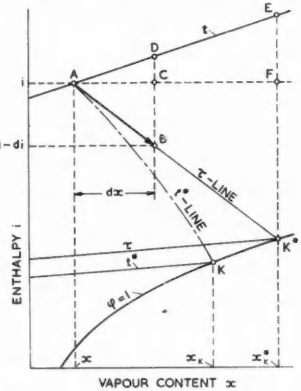


Fig. 8. **Diagram for derivation of conditions under which coinciding t^*-lines and τ-lines are obtained.**

In Fig. 8, the point A, corresponding to high i- and x-values, represents the state of air flowing under steady adiabatic conditions over a moist body with a temperature t^*. The state at the body surface is given by the point K corresponding to a vapour content x_K. Through A and K passes a t^*-line (shown chain-dotted) which represents constant terminal temperature. The true state of the air, however, varies along a τ-line through the point A, which intersects the saturation curve at the point K^*, which corresponds to a vapour content x_K^* and a temperature τ. Assume that, over an infinitely small distance, the state of the air varies a indicated by displacing the " air point " from A to point B. During this displacement, the enthalpy decreases from i to $i - di$ and the vapour content increases from x to $x + dx$. The points A, C, and F have the same enthalpy i, the points F and E have the same vapour content x_K^*, and the points A, D, and E correspond to the same temperature t. As a result,

$$DB/AC = EK^*/AF \quad .. \quad (19)$$

For the various segments, the relations are

$$DB = DC + CB = i_D\,dx - di$$

where $i_D = c_D\,t$ = enthalpy, and c_D = specific heat of the vapour, and

$$EK^* = c_K^*\,(t - \tau)$$

where $c_K^* = c_L + c_D\,x_K^*$, c_L being the specific heat of the air. Therefore, eq. (19) gives

$$(i_D\,dx - di)/dx = c_K^*\,(t - \tau)/(x_K^* - x)$$

so that finally

$$di/dx = i_D - [c_K^*\,(t - \tau)/(x_K^* - x)] \quad .. \quad (20)$$

According to eq. (3), the value of di/dx is also equal to $ct^* - r_0$, where $ct^* = i'$ is the enthalpy of the water. Denoting by i_0 the enthalpy of the vapour at the temperature t^*, it then follows that

$$r_0 + i_D = i' + r + i_D - i_0.$$

Therefore,

$$i' - r_0 = i_0 - r = di/dx = i_D - [c_K^*\,(t - \tau)/(x_K^* - x)] \quad (21)$$

and in conjunction with eq. (17), this gives

$$i_0 - r = i_0 - [(\alpha/\sigma)\,(t - t^*)/(x_K - x)] \quad (22)$$

The expressions on the left-hand sides of eqs. (21) and (22) are thus equal, with the result that

$$\frac{\alpha}{\sigma} = c_K^*\left(\frac{x_K - x}{x_K^* - x}\right)\left(\frac{t - \tau}{t - t^*}\right) - (i_D - i_0)\left(\frac{x_K - x}{t - t^*}\right) \quad (23)$$

If the t^*-line and the τ-line coincide, the relations $x_K = x_K^*$ and $t^* = \tau$ can be substituted in eq. (23). Also, since $i_D - i_0 = c_D\,(t - t^*)$, equation (23) reduces to

$$\alpha/\sigma = c_L + c_D\,x \quad .. \quad (24)$$

This final equation states that the τ-lines of equal cooling limit and the t^*-lines of equal terminal temperature will coincide and become a single straight line if the ratio of the transfer coefficients is equal to the specific heat of a moist gas which contains one unit weight of pure dry gas. With regard to Lewis's formula, the following may be concluded :—If the specific heat of a vapour and gas mixture is based on that of a mixture containing one unit weight of pure gas, then Le equals unity, when the τ-line and the t^*-line coincide for the same surface temperature of the body.

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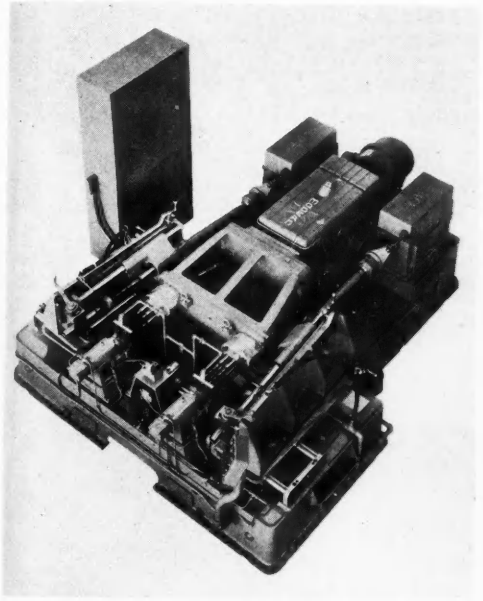
New Materials, Processes and Equipment

SPECIAL-PURPOSE DRILLING MACHINE

A special-purpose drilling machine, embodying a number of interesting design features, has recently been completed by Modern Tool Works Limited, of Toronto, Canada.

The machine has been engineered and built in close co-operation with Hayes Steel Products Limited, of Merritton, Ontario, as part of an equipment replacement programme, designed to reduce handling and to increase efficiency in the production of "I" beam front truck axles.

The design of "I" beam front axles requires certain critical relationships to be maintained between king-pin holes and the holes in the spring seat pads for centre bolts and spring "U" bolts. The king-pin holes and the spring seat holes must be located symmetrically about the centre line of the axle; at the same time, the king-pin holes must be drilled in the centre of the bosses provided on the ends of the forgings. Since the forgings, after straightening, may vary as much as $\frac{1}{4}$ in. between the centres of the bosses, there exists the problem of compensation when all holes are to be drilled simultaneously.



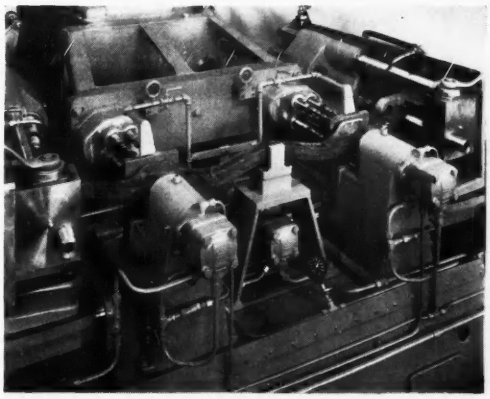
The new machine embodies the flexibility required to accommodate all the various truck axle designs encountered and features automatic compensation for variation occurring between like forgings. To overcome the problem of maintaining the king-pin holes in the centre of the bosses, regardless of the distance between them, the machine automatically adjusts the drilling spindles to suit the forging, while simultaneously it centralises the forging to the spindles which drill the spring seat pads, thus producing the two required conditions of relationship in one automatic setting.

The machine is designed around three "Modern" standard hydraulic feed units. The largest unit drills the holes for the spring centre bolts and spring "U" bolt holes, while the two outer units drill the king-pin holes. Feed rate is infinitely variable for different drilling conditions, and the outer units have additional provision for skip-feeding, required to maintain maximum production, when drilling king-pin holes in axles having clevis-type ends.

Spindle speed control is obtained by change gears, which are conveniently housed in a storage compartment in the base of the machine. For maximum ease of maintenance and set-up change-over, and to afford clearance for overhead handling of forgings, the machine is arranged horizontally and offers a comfortable operating height, which reduces operator fatigue to a minimum. Finished, drilled axles can be produced at the rate of 32 per hour at 80% efficiency.

The compensating device which positions the axle forging centrally and adjusts the drilling spindles for king-pin holes consists of two movable assemblies, which carry the drilling spindles, spindle slides, drill bushings, thrust support blocks, and "V" block locators in unvarying relationship. The "V" blocks are essentially the jaws of a self-centering vise mechanism, and at the same time serve as gauging locators to position the drilling spindles to intersect the forging bosses centrally.

The two moving-drill spindle and locator assemblies are operated by opposed-pitch leadscrews driven by a single hydraulic motor. The low pitch angle of the leadscrews produces a mechanical locking of the moving assembly at any point over the entire length of the screw, eliminating any necessity for adjustment for axles of different length. Pressures developed by the clamps are controlled by adjusting the hydraulic pressure, to



avoid distorting the axle beam. Similarly, the vises which grip the axle at the spring seat pads are screw operated by a hydraulic motor.

The combination of screw operation to the clamps and hydraulic drive produces two distinct advantages. A locking action over the entire length of the screw accommodates a great variation in size of parts and produces mechanical strength to contend with the powerful forces developed by the feeding of the drills, these approaching a total in some cases of 12,000 lb thrust, when twelve drills are cutting. The fact that the screws are hydraulically powered makes it possible to co-ordinate the function of the different clamping

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movements by automatic sequence control, thus removing the onus from the operator of maintaining a fatiguing routine of correct order, vital to the proper functioning of the machine.

To accommodate variation between axles of different design, the spindle and locator slides are adjusted manually in two planes, and are locked to produce the correct combination of caster and camber angles for the king-pin holes. Drive to the spindle is through a propeller shaft, and feed thrust is transmitted through a non-revolving propeller shaft housing.

The cluster heads for spring seat drilling are of the fixed centre type and are permanently mounted in pairs on adapter plates in correct relation for a given axle design. This system permits quick disengagement from the mounting flange and release of universal drive couplings, to facilitate frequent alteration of set-up required when producing short-run requirements.

The principal features of the machine can be clearly seen in the accompanying high-angle illustration, while the second illustration shows a close-up of an axle forging in the machine, before clamping. In the upper right-hand corner can be seen the drill spindle guides, bushing, and thrust support.

ELECTRIC PUMPING SETS

The latest addition to the extensive range of centrifugal pumps manufactured by Messrs. Sigmund Pumps Limited, of Team Valley, Gateshead, is the Sigmund "PumPak," which introduces a new standard in the design and performance of close-coupled electric pumping sets.

The two principal features which distinguish the PumPak from all other close-coupled pumping sets are the Sigmund stator pack service, and the special ventilation arrangement. In the usual type of integral pump, the difficulty of replacing the stator in case of electrical breakdown is often the main objection to the application of a close-coupled unit. The Sigmund patented method of withdrawing the stator, and the special stator pack replacement service, completely obviate this objection. The stator housing is so designed that it can be removed complete with stator without disturbing the pump in any way, all that is necessary being to disconnect the leads to the terminal box and remove four nuts, to enable the whole unit to be completely withdrawn. Stocks of each size of PumPak stators are held at the Works, so that in the event of an electrical failure developing in the set, a replacement stator can be despatched promptly, thereby enabling the pump to resume duty with the minimum of delay, inconvenience, and expense. The withdrawal and replacement of the PumPak stator can be carried out in a matter of minutes by comparatively unskilled labour, as the form of construction obviates any possibility of misalignment.

The importance of adequate ventilation has been specially studied and the cooling arrangement is another Sigmund patented feature. Cool air is drawn through the entire motor by a fan mounted on the pump side of the stator ; a stream of cool air circulates around the pump side bearing, effectively cooling the bearing, and preventing the entry of any liquid into the bearing housing.

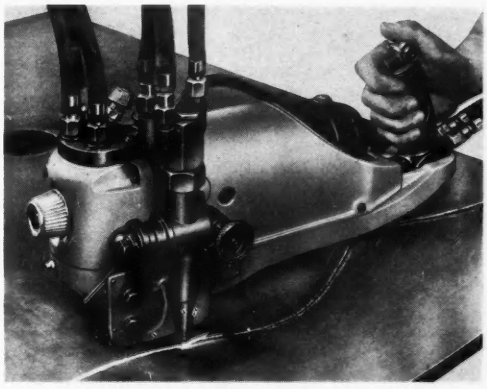
The initial PumPak range now in full scale production covers units from 1½ to 15 hp at 1450 and 2900 rpm, with capacities up to 500 gpm and heads up to 175 feet, giving a wide selection of units, but an additional series of fan-cooled and small types will shortly be available, enabling potential users to meet any specific requirement.

The PumPak operates efficiently in any desired position, is simple to instal, and gives an excellent all-round performance at low upkeep cost. Its extreme adaptability enables it to be advantageously applied to many service applications in many fields.

CUTTING MACHINE

The "Bantam", for straight-line and circle cutting, is the latest cutting machine from the British Oxygen Company Limited, of London, S.W.1. It is also the lightest, weighing only 22¾ lb including cutter, and, although small, it is one of the sturdiest machines of its kind ever produced, with many of the advantages of much larger and more expensive equipment.

With this new machine, mild steel up to 2 in. thick and, when used in conjunction with the radius bar, circles of between 3 in. and 45 in. diameter, can be cut. For bevelled edges, the cutting head is adjustable, enabling an inclined cut to be made at any angle up to 45 deg. It is also adjustable for lateral movement and height.



The cutter uses the "Cutogen" one-piece nozzle, which gives clean machine-finished cuts on slow curves, when free cutting. For this purpose the Bantam is steered by a handle at the rear of the machine and it can then be made to follow curves marked upon the plate surface, as shown in the illustration. The caster trailing wheel can be locked for running on a track and can be disengaged for circle and profile cutting by the simple adjustment of a locating screw. A 3 ft light-alloy track is provided with the machine, and additional lengths can be purchased separately for extending it.

There is a reversing switch for straight cutting, when mounted on the track and, to enable the nozzle to be quickly lined up over the work, a cross traverse arm is fitted with a quick-acting locking lever—a refinement not usually found on a machine of this kind. Performance depends, of course, on several factors, but as an indication of the machine's capabilities, in straight-line cutting, ⅛ in. plate can be cut at 130 ft/hr, ½ in. plate at 80 ft/hr, and 2 in. plate at 40 ft/hr, with the appropriate nozzle. In bevel cutting (up to 45 deg.), these speeds are 66, 42, and 30 ft/hr respectively.

All steel parts are oxidised to prevent rusting, light-alloy parts are anodised and attachments are chromium plated.

A general view of the Tool Room

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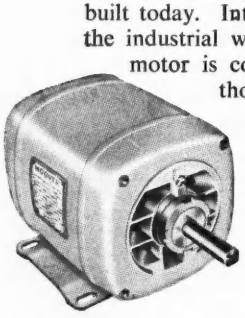

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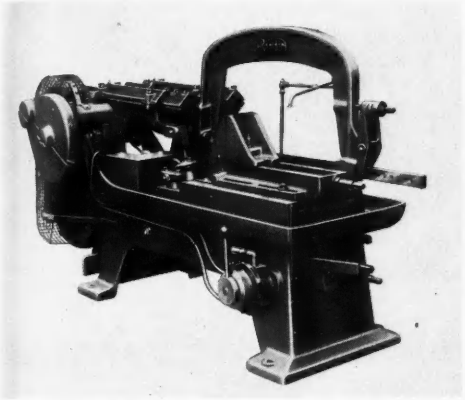
IMPROVED HACKSAWING MACHINE

Incorporated as a new feature of the 10 × 10 in. and 12 × 12 in. capacity "Rapidor" hacksawing machines manufactured by Edward G. Herbert, Ltd., of Levenshulme, Manchester, is a hydraulic pump unit, which raises the saw frame at the end of the cut, and also holds the saw frame at any convenient height above the work when inserting bars or sections which are smaller than the full capacity of the machine.

The new unit consists of a two-throw plunger pump, operated by two eccentrics on the mainshaft, a lever-operated control valve, an oil reservoir, and a new valve release incorporated in the dashpot piston. The arrangement does not interfere with the action of the well-known "Rapidor" automatic lift to the saw blade on the return stroke, but adjustment of the feed automatically controls the amount of this lift.



In action, oil is pumped under pressure from the reservoir to the control valve and, according to the position of this valve, the oil is either circulated back to the reservoir or fed into the dashpot cylinder below the piston, which is then forced upwards, thereby lifting the saw frame. On attaining the maximum height, the lift is stopped automatically by the action of a relief valve on the dashpot piston.

When the control lever is placed in the "Feed" position, the oil flows back to the reservoir, releasing the pressure on the dashpot piston, and thereby lowering the saw on to the work. There is also a "Dwell" position and, with the control lever in this position, sufficient oil is pumped into the dashpot to hold the saw frame at any convenient height, short of the maximum, while the operator fixes the work in the vice.

All parts of the new unit are designed as exterior fittings, so that they can be attached without difficulty to existing machines arranged for the standard single or multi-speed motor drive.

Tests undertaken with the unit show considerable improvement in cutting times on many materials.

MACHINE STOPPAGE RECORDER

A machine stoppage recorder, which shows not only when, but also why, such stoppages occur, is the "Indicator" Model TM Servis Recorder, manufactured by Servis Recorders Ltd., of Gloucester.

Whenever a machine stoppage occurs, the plunger-dial of the instrument is turned by the operator to a pre-arranged code number, and depressed. This causes a corresponding number of small holes to be punched by a row of needles in a chart at the appropriate position on a time scale. When the chart is later inspected, the cause of each stoppage is thus already indicated by number, thereby saving the time required for the investigation of stoppages and proving that the operator was attending the machine at the time when each stoppage occurred. Up to six different causes of stoppage can be indicated on the chart. Besides simplifying chart analysis and reducing clerical work, this instrument also gives plant engineers a clear indication of machine defects, often warning them of impending breakdowns.

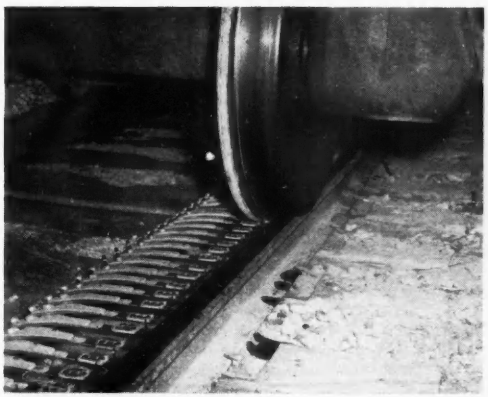
These recorders are available in 8, 12, 24, and 72-hour types, with or without clock face, as required.

RAILWAY WHEEL-CHECKING EQUIPMENT

Two American engineers, Messrs. W. A. and M. V. Gieskieng, have found what is claimed to be a purely mechanical solution to the problem of *en-route* railway wheel inspection. The Gieskieng equipment, known as "Wheel Checkers," has been installed on the Ventura County Railway in California for detecting defective wheels on moving trains. It consists of two units, the first of which checks for broken flanges and loose wheels (out of gauge), and the second for broken treads.

A bell and a coloured light signal are connected to detectors through a relay and battery circuit. A train, moving over the detectors with a defective wheel, will complete the circuit by means of the relay and battery and will cause the bell to ring and the signal to display a red aspect. The bell can be shut off and the signal can be restored to normal by manual operation.

The broken-flange detector consists of "fingers," made of spring steel and mounted at right angles to the rail, together with an insulated stainless steel pad located near the end of each finger. The end of each finger is upturned and extends about an inch above the top of the rail. A guard rail, mounted opposite the fingers, acts as a guide for the wheels. When a normal



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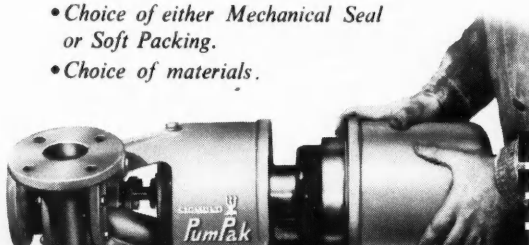

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wheel passes over the unit, its flanges come into contact with the insulated stainless steel pads, thus depressing the fingers away from the wheel; where, however, the flange is broken, the steel finger under the break is not depressed, and hence the upturned end of the finger makes contact with the wheel, completing the electrical circuit. Furthermore, owing to the fact that a loose wheel would fail to stay on the insulated steel pad, it also would be detected.

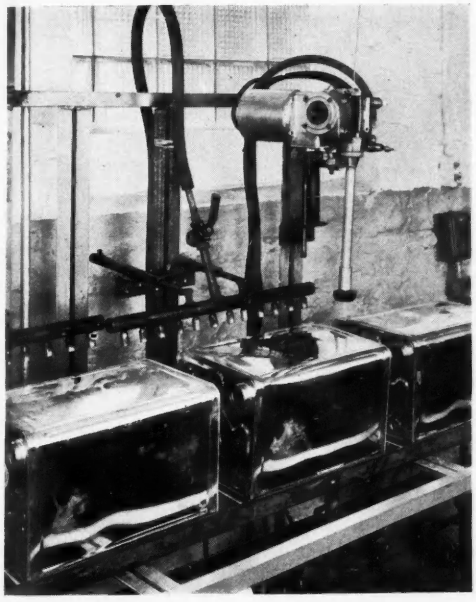
The unit, which is 12 ft in length, acts as a detector for wheels up to 42 in. in diameter, with a broken flange at least $3\frac{1}{2}$ in. in length and within $\frac{5}{8}$ in. of the tread.

The broken tread detector consists of a detector rail located slightly below the top of the other rails. If a wheel with a broken tread, running over this detector, drops $\frac{3}{16}$ in., the circuit is completed.

" Wheel Checkers " are designed to operate with a train moving at 15 mph, but with the use of an electronic relay and dampers, train speeds can be increased to 30 mph.

SOLDERING AND BRAZING GUN

A new kind of soldering and brazing gun, called the " Mogul," intended for high-speed and production seam soldering, has recently been developed by the Metallizing Company of America, of Chicago, Illinois.

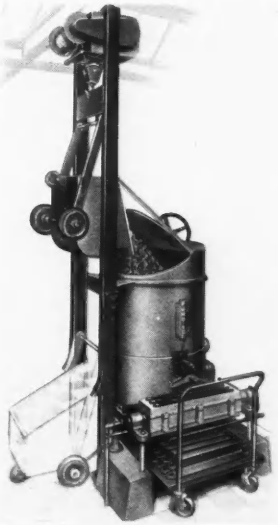


In actual operation, the workpiece travels past the gun nozzle at a constant speed and the amount of solder " dripping " or depositing is always that for which the gun is actually set. The motor speed is constant and the wire feed is varied by a variable speed control. The wire speed can be stopped and started without shutting off the flame. The soldering or brazing wire is melted approximately 90% by the flame of the gun and, in a semi-liquid form, makes perfect contact with the work-piece, which supplies the balance of the heat required to melt and flow. If required, solder can be deposited in a completely liquid form, controlled by the operator. It may be desirable in certain applications to drop the solder in a series of fine drops. If so, the size and the spacing of these can be thoroughly controlled.

The illustration shows a typical application of the gun.

AUTOMATIC LOADER FOR MELTING FURNACES

The latest development of Messrs. Funditor Limited, of London, E.C.1, is the Funditor " Auto-Loader," designed for the automatic loading of melting furnaces, with the consequent elimination of the hard work of man-handling metal into melting furnaces—a process known and accepted as costly in terms of man-hours.



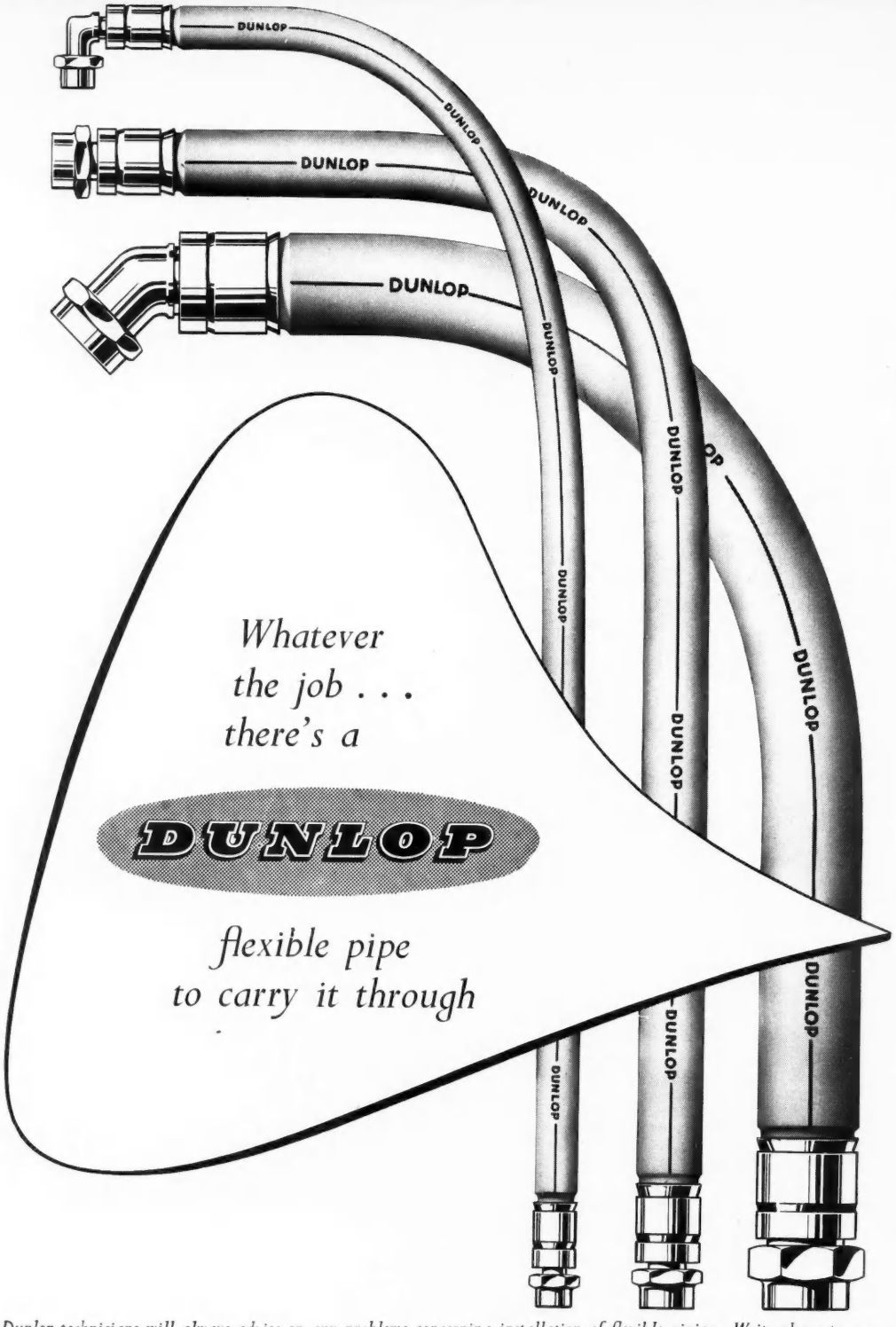
This automatic metal loader is inexpensive, is quickly installed, and is operated by push-button control. Once the loaded mobile skip has been wheeled into position, the motorised hoist lifts it to the requisite height and opens the furnace lid, to enable the skip to discharge its load into the furnace. The empty skip is then returned to floor level. The whole sequence of operations is semi-automatic and the operator's action is confined to pressing a push-button to raise the skip, and a second push-button to lower it.

The illustration shows the " Auto-Loader " working in conjunction with a 1-ton capacity Funditor furnace, but in most cases it can be applied, when suitably modified, to other types of melting furnaces.

ADJUSTABLE LEATHER V-BELTING

The " Adjusticon " adjustable leather V-belting, manufactured by Messrs. Carol Beltings Ltd., of London, W.C.1, is a multi-ply leather V-Belt, with flush unbroken driving faces, giving absolute grip, flexibility, and resilience. It is reinforced throughout its entire length by stainless, evenly spaced tubular metal eyelets, which serve to clamp the plies of the belt, preventing their shearing, and effectively air-cooling the belt, when in circulation. By means of these eyelets and the specially constructed " Adjusticon " fastener, the belt can be easily adjusted for length and tension on any V-belt drive. The fastener is held in place by a steel spring plate, and because of its very short overall length and the belt ends abutting within the fastener, maximum flexibility and resilience are maintained, as well as even and silent transmission, no matter what size of pulley is used. No metal parts come into contact with the pulley groove, and the locking device of the fastener can be released only by manipulation.

Owing to its simple construction and adjustability, it is suitable for all types of V-drives, especially difficult ones, such as " closed " drives and drives where stretch take-up and/or tension is an involved operation. It


DUNLOP
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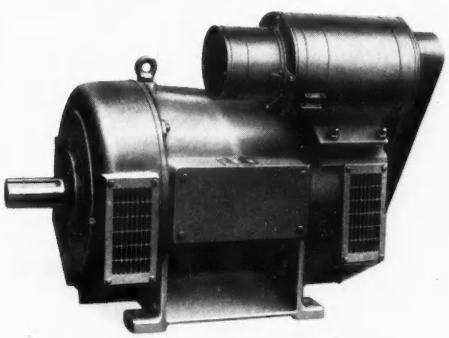
can even be applied with success in locations where endless V-belt transmission has hitherto been impossible. It compares favourably in price with all other V-belts, and, since it is cut from the coil to the required length, it is claimed to be the most economical belt for all V-drives.

"Adjusticon" V-belts are made in standard sizes and are supplied in coils of 25, 50 or 100 ft, with the respective sizes of fasteners.

SELF-REGULATING ALTERNATORS

The self-regulating alternators, with "Amplidex" control system, produced by The Brush Electrical Engineering Co. Ltd., of Loughborough, have been developed to meet the demand for machines capable of operating continuously with the minimum of maintenance. The "Amplidex" control system is compact and static, and, as a result, setting-up difficulties are obviated and the machines are ideally suited for working in unattended situations.

One virtue of the "Amplidex" control system is that it can be applied to an alternator of standard and well proved construction. The alternator has therefore the normal salient pole construction, with large air gaps and normal proportions of field and armature ampere turns, just as if it were required for use with an ordinary automatic voltage regulator. Consequently, it has the same capacity for dealing with leading power factor loads. "Amplidex" control is achieved with an amplifying dynamic excitation unit and a simple voltage detection circuit. The alternator voltage is fed through an isolating transformer to two bridge rectifiers, one shunted by a saturated reactor and the other with a series capacitor. The outputs from the rectifiers are applied to two resistances in series and the difference in voltage across these two resistances supplies the error signal to the sub-exciter. A small rheostat is provided for setting the alternator voltage. The saturated reactor holds the voltage constant across one rectifier, but the voltage of the other varies in direct proportion to the alternator voltage. When the two voltages are equal, no current will be supplied to the exciter field. If the alternator voltage varies, the exciter field current will vary in magnitude, depending on the strength and direction of error signal, thereby rapidly correcting the terminal voltage to its set value. The reference voltage provided by the saturated reactor is sensitive to frequency, but so also is the current in the capacitor fed by the rectifier. These two effects are thereby cancelled out, and thus no variation due to frequency changes results. Over the normal range of engine governing, the balance point will remain unchanged; in fact, the frequency can be adjusted from 50 to 60 cycles with only a small change in voltage.



Brush self-regulating alternators with "Amplidex" control give a voltage regulation of $\pm 2\frac{1}{2}\%$ from cold to hot—at variable power factors from no load to full load—allowing for the usual 4% engine speed variation. When full load at 0·8 power factor is thrown off, the voltage rise is arrested in less than 0·2 sec, and the voltage overshoot is within 20% of normal voltage. Throwing off full-load unity power factor is less onerous and very little disturbance of the voltage occurs. Conditions for throwing on load are similar and voltage variations of the same order occur. Within the limit of full rated kVA, voltage recovery is obtained in one second or less for any condition of load change. These self-regulating alternators are capable of taking heavy load surges, with a minimum voltage disturbance and without loss of control.

This system will operate over a wide voltage range, such as 380/440 volts and a frequency range of 50/60 cycles. If necessary, a series/parallel stator connection may be provided to give dual voltage, when required. In this case, the connections to the control circuit remain unchanged, so that both voltage ranges are covered by the same rheostat. This scheme permits delivery of full rated output at either voltage.

Brush self-regulating alternators will operate satisfactorily in parallel, or with alternators controlled by conventional automatic voltage regulators, and can be supplied with outputs up to 150 kW.

PHOTOELECTRIC DUST METER

Although measurement of duct concentration in steel works by gravimetric methods is accurate, it must necessarily take place after passage of the particles through the plant. Thus, it is impossible to determine by this means whether a uniform dust concentration has existed over the set measuring period or whether the deposit has been caused by a few short periods of high dust concentration from intermittent failure or overloading of some sections of the cleaning plant. This has been overcome by the design of a continuously recording instrument, manufactured by Radiovisor Parent Ltd. of London, N.W.1, which monitors the gases at any chosen point in the main; it may be calibrated in conjunction with standard gravimetric measurements, and to suit site conditions. The equipment enables an alarm to be sounded immediately a pre-determined dust level is exceeded, and it also provides a continuous record of the dust concentration and its variations.

The instrument, known as the BISRA photoelectric dust meter, comprises four main components, *viz.*: The sampling chamber, in which are installed photoelectric cells, the light source, and an optical system; a thermionic valve amplifier, together with its stabiliser, power supplies, and setting-up controls; a 7-in. diameter indicator-relay meter, incorporating a small contactor for connection to an alarm system; and a continuous $3\frac{1}{2}$-in. wide strip recorder.

The light source and optical system in the sampling chamber give a parallel beam of light of uniform intensity, which is projected through the actual viewing chamber on to a light trap, specially designed to prevent secondary light radiation. The viewing chamber consists of a series of baffles to maintain the cylinder of light, and a glass cylinder with gas-tight joints; on the outside of the cylinder are three photo-multiplier

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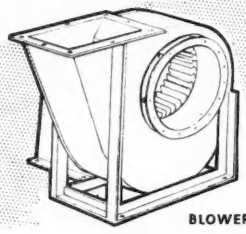

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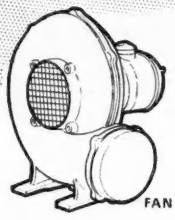

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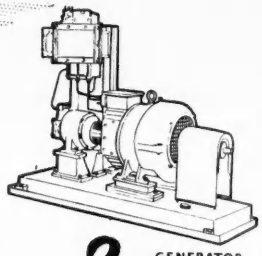

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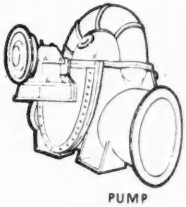

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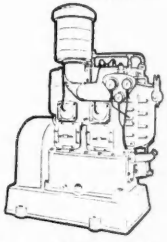

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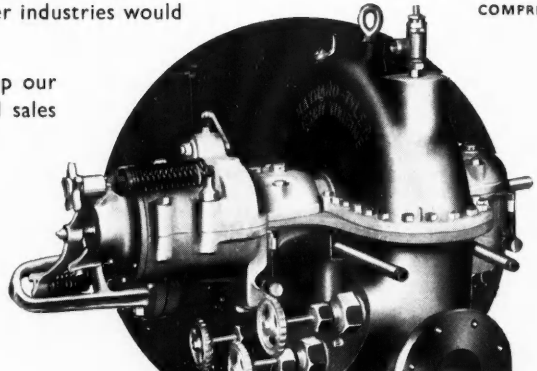

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CLASSIFIED ABSTRACTS

Numerous important abstracts which, for lack of space, cannot be published in our main editorial columns are regularly included in condensed form in this section. Subscribers may obtain photostat copies of all original articles at cost.

ELECTRONICS

Particle Counting by Television Techniques.

By L. E. FLORY and W. S. PIKE. (From *RCA Review, U.S.A.*, Vol. 14, No. 4, December 1953, pp. 546-556, 9 illustrations.)

THE problem of counting the number of microscopic particles in a given area appears to be important in many scientific fields. At present, the counting of such particles as blood cells, bacterial cultures, grains of photographic emulsions, etc., is a laborious process, as a technician must view the field of particles in a microscope and count them visually with the aid of a mechanical counter or similar device. There are, in addition, many chances of error in such a process, as particles may easily be overlooked or counted more than once.

Provided the particles are regular in size and are not clustered too closely together, television scanning techniques offer considerable promise for rapid and accurate counting. This paper describes such a technique. The particles to be counted are placed under the microscope and are suitably illuminated. The camera of a closed-circuit television system is substituted for the eye of the observer at the microscope eyepiece, and focus and illumination are adjusted by observation of the monitor kinescope of the television system. The video output of the latter is applied to a count-rate meter. The particles to be counted will produce video impulses at the output of the system, and the count-rate meter will give an indication proportional to both the number and size of the particles in the microscope field. To make the reading independent of particle diameter, compensation is applied.

The first experimental application of this technique has been in the counting of red blood cells, or erythrocytes. For convenience, early testing was accomplished with simulated fields of particles. The television camera and a slide projector were set up side by side and focused on a screen. A microscope was not used. Slides, simulating the appearance of fields of particles, were made by punching appropriate sizes and numbers of holes in pieces of opaque paper, and these were then sandwiched between glass plates and inserted in the slide carrier of the projector. By suitable adjustment of the relative positions of screen, camera, and projector, a range of pulse widths and numbers of particles, entirely comparable with that obtained during actual erythrocyte counting, could be obtained. The results of these tests showed that good agreement between the electronic counter readings and the actual number of simulated particles was obtained over about a two-to-one range of particle diameters. The electronic count, with few exceptions, is within $\pm 5\%$ of the true number of simulated particles.

Although the present instrument has so far been used on erythrocytes only, it is concluded that rapid electronic counting of microscopic particles, with sufficient accuracy for many purposes, is technically feasible with this technique. The general application of the system to such fields as routine erythrocyte counting probably depends more on economic factors than on technical possibilities, but, with the increasing availability of inexpensive, closed-circuit television chains, it is possible that widespread application of this technique may take place.

FORGING PRACTICE

Cold "Rocking" of Tubing.

By K. W. DALE. (From *The Iron Age, U.S.A.*, Vol. 172, No. 19, November 5, 1953, pp. 172-174, 8 illustrations.)

RECENT experience in the cold compression forming of tubing and tube forms by "rocking" has opened a wider field for application of this method to modern part design problems. Compression-formed tubing differs basically from cold-drawn tubing. Tubing produced by the Tube Reducing Corporation, Wallington, N.J., by the "Rockrite" process, is essentially a cold-forged product.

Properties of cold-worked steels are affected by the process used. In cold-drawn tubing, the metal is in tension. Maximum reduction per pass, about 40 per cent, must be held within limits of the tensile strength of the material or the tube will pull apart. In rocked tubing, reduction is limited only by the ultimate compressive strength of the material. Single pass reductions in excess of 85 per cent can be made on carbon and stainless steel tubing.

In the rocking operation, dies roll back and forth over a stationary tube. The dies have a tapered groove around half their periphery. The groove at one end is slightly larger than the outside diameter of the tube to be sized. The groove narrows, and at the other end of the die is the proper size for a desired tube diameter.

The mandrel, against which the tube wall is squeezed, is also tapered. This permits the tube, as it elongates, to feed free of the mandrel. As the dies roll along the tube, the narrowing die grooves compress the tube walls against the mandrel. Outer and inner diameters and wall thickness are reduced; tube length is increased. The tube is intermittently fed forward through the dies and rotated about one-sixth of a turn. Alternate feeding and indexing spread the compressive cold working evenly over the entire surface of the tube.

The effects of this method of forming are reflected in the changed microstructure of the metal. A finer grain is developed, while the large reduction per pass substantially thins out the decarburising layer, and minimum machining is needed to remove the decarburised skin remaining. This method of cold forming develops mechanical properties considerably greater than those normally produced by cold drawing. Furthermore, the relatively small amount of motion between the die and the tube has made practical the cold reduction of many metals, normally considered hard to draw.

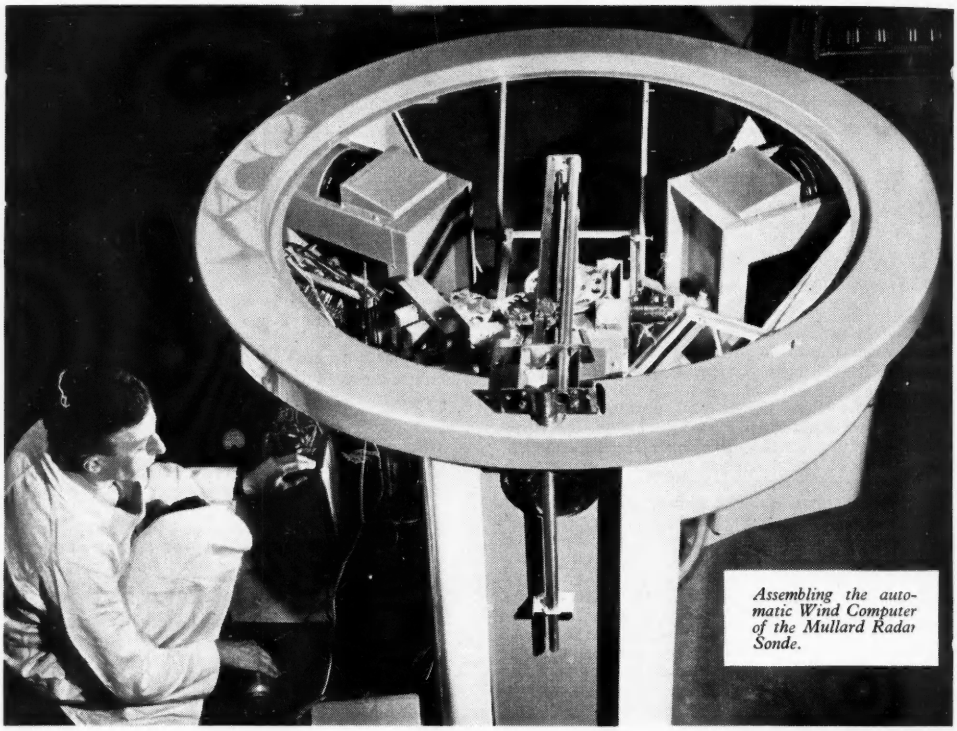
Both inside and outside surfaces of rocked tubing have a fine finish. A slight amount of controlled slippage between die and tube burnishes the outside, while repeated striking of the inside surface against the polished mandrel produces a planishing action.

INTERNAL COMBUSTION ENGINES

The Effect of Fuel Addition to Intake Air on the Performance of a Compression Ignition Engine.

By W. T. LYN. (From a paper presented at a meeting of *The Institution of Mechanical Engineers, London*, January 8, 1954, 24 pages, 31 illustrttions.)

THIS experimental investigation into the effects of fuel addition to the intake air of a compression ignition



Assembling the automatic Wind Computer of the Mullard Radar Sonde.

PROGRESS IN ELECTRONICS

THE rapid growth of air traffic throughout the world and the introduction of high speed aircraft flying at great heights have multiplied the problems of the meteorologist. The methods of forecasting weather conditions which were adequate ten years ago cannot provide the detailed and accurate data which are required to-day.

A new technique for weather forecasting has, however, now been developed. By employing an equipment known as the Radar Sonde, the complex conditions in the upper atmosphere can be accurately measured and recorded.

A small balloon carries a miniature radio and meteorological station up to a height of 100,000 ft. and is borne by the wind to distances exceeding 100 miles. Throughout its flight the balloon is continuously interrogated by a powerful radar transmitter, and the answers are transmitted back by return.

As the information is received at the ground station it is decoded, computed and recorded to give a detailed picture of wind speed and direction, and of temperature, pressure and humidity.

The Radar Sonde has been designed and built by the Mullard Research Laboratories in co-operation with the Ministry of Supply and the Meteorological Office.

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engine arose from a programme of research into the means of reducing engine roughness and combustion noise. Information on pre-flame reactions in spark-ignition engines from published literature indicates that these reactions, which yield active partial products of combustion finally leading to the auto-ignition of the end gas, or "knock," may reduce the ignition delay in a compression-ignition engine and contribute to the smooth running of the engine.

Preliminary tests with part of the fuel introduced with the intake air in the form of spray confirmed such a view. They showed how, by slow oxidation of the aspirated fuel, the compression pressure is increased and the ignition delay of the main fuel is reduced, with consequent smoothing of the pressure diagram and elimination of combustion noise. The effects on combustion and performance under various loads and speeds are presented, it being found that overall thermal efficiency is improved at high load, but is made worse at low load.

Further investigations were then carried out on the various factors that may have effects on the result of introducing fuel with the intake air. Among these factors were the types of fuel used for the aspirated as well as the main charge, combustion chamber design, compression ratio, and timing of both parts of the charge. An analysis is made of the processes of combustion under such conditions and a theory advanced to account for the phenomena observed. It is concluded that the lean aspirated mixture does not proceed to complete combustion. The use of intake spray was found particularly effective when the main fuel was of low ignition quality.

The technique of using the slow oxidation of the aspirated fuel was then applied to study the ignition of various hydrocarbons in a compression-ignition engine. It is shown that the differences in ignition delay of various fuels lie chiefly in the chemical part of the delay, while the physical part remains essentially constant despite widely different physical properties of the fuel. It is shown also that with the use of intake spray, the compression-ignition engine can run on any fuel, irrespective of its cetane number and with only small differences in thermal efficiency.

Finally, the effect of some additives on the reactions of the aspirated fuel and the ignition of the main fuel was studied. The results showed that the reactions were suppressed by the addition of formaldehyde and tetraethyl lead and slightly accelerated by nitrogen peroxides. The indication was that the reaction is of the "low-temperature" mode. This is discussed in relation to recently published works on oxidation of hydrocarbons in general and the reactions leading to knock in the spark-ignition engine.

METALLURGICAL ENGINEERING

Metallurgical Aspects of High-Temperature and Gas-Turbine Plants.

By J. M. ROBERTSON. (From a paper presented at a meeting of the *North-East Coast Institution of Engineers and Shipbuilders, Newcastle-upon-Tyne*, January 15, 1954, pp. 217-252, 10 illustrations.)

IN connection with the properties of metals, it is convenient to describe a high temperature as one at which time has to be taken into account as well as stress in relation to the deformation and fracture of a particular metal. The ultimate tensile strength of mild steel is about 28 tsi at atmospheric temperature, and it will not break under appreciably lower tensile stress, no matter how long it is applied. At 900° F its ultimate tensile strength is about 16 tsi, but it will break under a tensile stress of 6 tsi if this is applied for 10,000 hours. Insofar as mild steel is concerned, atmospheric temperature is in the range of ordinary temperatures, and 900° F in the range of high temperatures, and the border line between ordinary temperatures and high temperatures is generally taken to be about 800° F. For many metals, such as those in which aluminium or copper are the principal constituents, the border between ordinary temperatures and high temperatures is well below 800° F; for others, such as many of those referred to in this paper, it is well above.

The fact that time has to be regarded as equivalent to stress, when a metal is used in a range of temperature which is high for that metal, makes the use of metals at high temperatures a more complicated matter than their use at atmospheric temperature. In addition to this, however, account has to be taken of the fact that when metals are used at high temperatures, constitutional and structural changes can occur in them while they are in service.

With regard to the selection of materials for components operating at high temperatures, information is required about mechanical behaviour in different conditions of treatment under different stress/temperature/time conditions, about resistance to corrosion by different environments, about ability to be cast, forged, rolled, or drawn to produce the components required, about the possibility of heat treating the component in order to develop the characteristic properties of the alloy, and about weldability, machinability, and costs.

In this paper, a brief account is given of the properties which are important in metals used at temperatures above 850° F, i.e., resistance to creep, rupture, high-temperature fatigue, relaxation, thermal shock, and high-temperature corrosion. This is followed by a discussion of the composition, treatment, and properties of the steels and other alloys available for high-temperature applications, i.e., non-stainless ferritic steels, stainless ferritic steels, austenitic steel, and alloys based on nickel and cobalt. The paper concludes with a section on welding, in which those aspects of welding which are particularly relevant to high-temperature applications are discussed.

RAILWAY ENGINEERING

The Wearing Parts of Electric Rolling Stock.

By A. W. MANSER. (From a paper presented at a meeting of *The Institution of Locomotive Engineers, London*, January 20, 1954, 42 pages, 34 illustrations.)

WHEN new rolling stock is introduced on any system, such factors as its physical design, load-carrying capacity, and performance are inevitably the subject of review in the technical journals and, although mention may be made of features intended to facilitate and reduce the cost of maintenance, the degree of success achieved by such features is rarely revealed, for the simple reason that it takes a long time to determine what saving in maintenance cost, if any, has been achieved.

With stock operating under the intensive conditions obtaining on the London Transport system, the author's experience is that the curve of maintenance cost rises year by year and does not show signs of becoming asymptotic until more than ten, and probably nearly fifteen, years have elapsed from the date of entry into service. In introducing changes in design intended to reduce maintenance costs one is, therefore, always dealing with a long-term project. Costing information going back over fifteen years is apt to be unreliable and firm comparisons are difficult to achieve. It is, however, apparent when a really worth-while change has been made, and this summary of experience with London Transport rolling stock has been made with the idea that it may be of value to others in showing design trends which have proved beneficial in reducing maintenance costs.

The wear to which all the component portions of vehicles are subjected may be divided into two categories, i.e., that resulting from doing the work for which they are provided, and that resulting from the vibration to

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which the vehicle in general, and the bogie frame in particular, is subjected. The latter type of wear is by far the more serious in many cases, particularly with such components as brake levers, which are mounted on the bogie frame and which have less than half the springing of the vehicle as a whole.

A large part of this paper is devoted to the wearing parts of bogies. The experience of wear quoted is in all cases that which has been observed with the stock on the London Transport system and, although generally this represents the type of wear experienced on other systems, there are cases, e.g., tyres, in which the wear is quite different from that to be found on a normal main line or inter-urban system, as it is peculiar to a track layout having many curves and many of them of far too small a radius. The system is, however, sufficiently extensive (248 route miles) and the number of vehicles sufficiently large (4050) for representative impressions as to wear-and-tear to be obtained. These vehicles operate between them some 585,000 car miles each day. Amongst other equipment operations which run into high figures, may be mentioned pneumatic doors, totalling four million operations per day and necessitating a high standard of reliability.

SCIENTIFIC INSTRUMENTS

Salinity Meter for Sonar Measurements.

By K. E. HARWELL and D. W. HOOD. (From *Electronics, U.S.A.*, Vol. 27, No. 1, January 1954, pp. 172-174, 5 illustrations.)

PRECISE measurement of the salinity of sea water is necessary for calculation of refraction gradients and sound velocities in sonar operations, in determinations of dynamic heights for ocean current predictions, and as an aid in biological, chemical, and geological studies.

At constant temperatures the conductivity of sea water varies rapidly with salinity and can be used to indicate salinity accurately. Instruments for measuring conductivity have been built but, as they operated at audio frequencies, it was necessary to have the electrodes in direct contact with the water sample. Fouling and polarisation difficulties have prevented these instruments from attaining widespread use. The use of r.f. current for conductivity measurements, however, has the advantage that the electrodes need not be placed in direct contact with the water, thus eliminating these difficulties.

In a new salinity meter, conductivity is measured in a glass tube of about 10 mm inside diameter and 120 mm in length. A band of metal about 20 mm in length is fused to the outside of the tube near each end to form electrodes. A 14-Mc signal from a stabilised source is passed across the electrodes through the solution and is then rectified with a crystal rectifier. The resulting d.c. voltage provides an indication of the salinity of the sample.

Conductivity of sea water changes rapidly with temperature, approximately 2·1% per °C, while salinity is practically unaffected. Because of this large coefficient, it is necessary to measure the temperature of the sample with an accuracy comparable with that of the conductivity measurements. A change in temperature of 1·34° C causes a change in conductivity equivalent to a change of one part per thousand in salinity in the high-salinity range. As it is often desired to measure salinity with an accuracy of 0·01 to 0·05 parts per thousand, the temperature must be measured to approximately 0·01° C.

The temperature of the sample in the present instrument is measured with a mercury thermometer inside the cell case, visible through a narrow, plastic window. To avoid temperature changes caused by heat absorption while the sample is in the cell, the cell case is lined with plastic foam insulation. A container of silica gel mounted in the case dehydrates the air, to eliminate condensation on the glass tube between the electrodes.

In operation, a sample of water having a salinity near the maximum value to be measured is placed in the measuring tube and a potentiometer is adjusted to an arbitrary high-scale reading by changing the potentiometer d.c. voltage or by readjusting the r.f. coupling coil. A standard impedance is then switched into the measuring circuit to replace the sample tube and is adjusted to give an arbitrary reading near the high end of the scale. The standard is not changed during any given series of runs, but is frequently switched into the circuit to check scale reading for drift or other errors. The potentiometer is adjusted to give the same reading each time the standard impedance is in the circuit, establishing a fixed relation between scale reading and conductivity. This instrument is adequate for a large percentage of the salinity measurements ordinarily made, although its accuracy of ± 0·04 per mil is somewhat less than that obtainable by chemical analysis. The accuracy of the instrument seems to be limited by the stability of the r.f. output voltage, accuracy of temperature measurements, and instability of the germanium diodes used. Because of these limitations, it is doubtful whether the accuracy can be very greatly increased.

STRENGTH OF MATERIALS

Limit Design of Columns.

By P. C. HARRIS. (From *Journal of the Aeronautical Sciences, U.S.A.*, Vol. 21, No. 1, January 1954, pp. 43-49, 16 illustrations.)

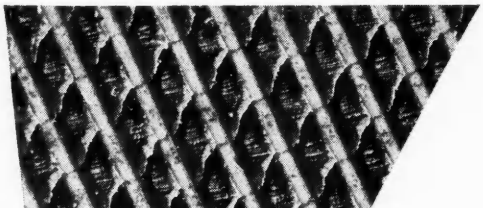
IN much of the literature of strength analysis, considerable emphasis has been placed on the maximum axial load a column will withstand. This load will usually be the controlling factor in a statically determinant structure, where failure of one member means failure of the whole structure. However, in a redundant structure, a column may or may not be at its maximum load at the instant the maximum load for the entire structure is reached. The axial deformation of the column may be beyond the point of maximum load or may not have reached it at all on its load-deformation curve.

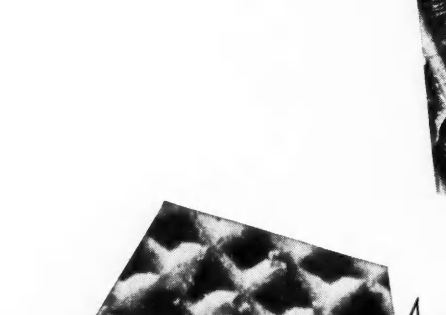
To formulate an analysis of structures, in which this fact can be taken into account, it is necessary to derive the load-deformation relationship for each member of the structure. From these individual member relationships, an overall load-deformation relationship for the entire structure can be found. The maximum value indicated by this overall relationship will be the true criterion of the strength of the entire structure.

In this paper, an approximate load-axial deformation relationship is derived for pin-end columns well into the inelastic range. A series of tests for length of member/radius of gyration ratios from 34 to 159 are shown for correlation with this approximation. The concept of " plastic-dynamic buckling " is introduced to emphasise the strain-rate energy dissipation in early plastic buckling of columns. This physical concept is applied to a generalised structure whose maximum load-carrying capacity is reached in the inelastic range.

Recently, emphasis has been placed on blast analysis of structures. A blast may cause sudden large deformations in structures. The energy of the blast dissipated in the structure will be partly dissipated in the compression members. The theory presented in this discussion affords a means of considering the large deformations and also the energy dissipation in columns, due to such dynamic loading. It is worth noting that equivalent length concepts permit this discussion to be extended to include columns with fixed ends. Similar treatment may be given to beam columns.

In short, the paper details a new concept of failure of redundant structures containing columns. This concept may, in part, explain many of the instances where





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testing has indicated over-design. This new theory will be applicable in cases where minimum weight and strength are primary considerations, although limitations of use are realized, due to large deformations.

VENTILATION

Natural Ventilation.

By E. YOULD. (From *Journal of the Institution of Heating and Ventilating Engineers, London*, Vol. 21, No. 221, January 1954, pp. 357-371, 10 illustrations.)

THE widespread employment of natural-ventilation systems in industry is itself the best advocate of the possibilities of correctly engineered ventilation schemes, utilising natural forces as the motive power. The main advantages of natural systems are featured in relatively low installation charges, low maintenance costs, and the absence of running charges. In addition, natural systems are silent in operation and can function out of working hours without attention.

Natural ventilation results from the effects of the wind and temperature difference between the air inside and outside a building, and, because wind varies both in direction and strength, it is a prime requirement that outlets should be designed so that the wind does not come into opposition to the natural outflow due to temperature difference. If to this can be added the feature of the wind producing a suction at the outlet, so much the better.

It is this necessity for harnessing the effects of wind which has had a profound influence on the development of natural exhaust equipment, and it is now generally accepted that a natural scheme must consist of a system of suitably designed inlets at a low level, coupled with outlets on the roof with only a positive reaction to the wind from any quarter. Cross ventilation through openings in vertical areas at different levels is seldom considered, except for domestic, office, and small factory buildings with no especial ventilation problem. It is inherent in such schemes that a high degree of control is necessary, for openings which are outlets one day, may be inlets the next day, if the wind changes.

Though dependent on forces of wind and temperature difference, inherently variable, it is possible by correct design and installation of equipment, and reasonable appreciation of atmospheric conditions, at times when ventilation is most needed, to overcome adverse tendencies often quoted in condemnation.

This paper recognises that the effect of changing wind velocity and direction is the major problem for solution, and emphasises that schemes designed to operate only during prevailing wind conditions may prove unsuitable where the need for ventilation is constant and, where adverse winds may persist for weeks on end, as in Great Britain.

Correctly proportioned systems of outlet and inlet can deal with, besides general ventilation, problems of heat, steam, and fume removal, and drying processes, but none of these applications is likely to respond to a haphazard approach.

VIBRATIONS

The Graphical Solution of Transient Vibration Problems.

By R. E. D. BISHOP. (From a paper presented at a meeting of *The Institution of Mechanical Engineers, London*, January 22, 1954, 16 pages, 26 illustrations.)

WHEN a vibration analysis of a mechanical system is made, it is first necessary to simplify the problem before applying to it the rules of mechanics. In this way, it is very often possible to arrive at a simple system of a single degree of freedom, where a mass is attached to a linear ("Hooke's Law") spring and is subjected to some time-dependent applied force.

It is clear that, when accuracy of working is important and a computing machine is available, numerical methods have much to recommend them. But graphical constructions may sometimes be more useful because they give information of a more general character than do their numerical counterparts. Thus, they appear to be the more desirable for giving instruction on the effects of transient forcing. The present paper is devoted to a graphical method involving use of the "phase-plane," i.e., it is proposed to discuss the behaviour of the simple oscillator by means of a plot of $\dot{x}$ against x (or, what is dimensionally more desirable, $\dot{x}/\sqrt{(k/m)}$ against x). Phase-plane methods have been used with great success in the theory of non-linear mechanics. They have also been applied to a different class of problems, *viz.*: Dynamical systems with transient loading. The paper concerns this second use of the plane and some recent extensions of it.

A survey is made of the literature covering the twenty years since these constructions were introduced. This relates to the transient disturbance of single-degree-of-freedom systems (which may include damping, non-linearity, hysteresis, etc.) and of multi-degree-of-freedom systems; the deflections of beam columns in the theory of elastic stability; and Jacobsen's general method of graphical integration.

Some new additions to the theory are given. These are: (1) A method of graphical differentiation; (2) a simplified approach to multi-degree-of-freedom systems leading to the treatment of transient loading of beams by moving and moving-and-varying loads (damping may be contemplated in these beam problems and the method devised easily carries over to certain plate problems); and (3) a demonstration that the graphical method can be used for deflections—as well as bending moments—in the theory of beam columns.

It is suggested that these constructions are of real importance, not only for practical engineering problems, but also for teaching purposes.

NEW MATERIALS, PROCESSES AND EQUIPMENT

(*Concluded from page* 75)

cells. The complete unit is suitably insulated to reduce heat losses.

The instrument works on the "light-scattering" technique. Thus, by introducing clean air into the viewing chamber, a minimum of light scatter takes place, and a minimum current is passed by the photocells. The output from the electronic amplifier is "backed off" so that, in this condition, a zero reading is shown on the recorder and meter. Any increase in light scatter due to dust particles suspended in the gas results in an increase in the photo-multiplier current, with a corresponding deflection of the needle of the indicator meter.

In use, the blast-furnace gas is taken from the main, raised above dew point by preheating it with steam coils, and fed through the viewing chamber. After recording the dust concentration, it is then either injected back into the gas main by an impeller, or exhausted to atmosphere. To prevent aggregation of the particles, the design of the sampling tube incorporates smooth bends, gate-type valves, and a thin-walled receiving tube. Gas velocities in the main stream are measured, and the flow rate in the sampling by-pass controlled, by an orifice plate; this ensures that a correct comparative reading of the dust concentration is obtained.

The equipment has undergone prolonged trials under the guidance of the British Iron and Steel Research Association at various steel works, and the figures obtained have proved consistent when checked by gravimetric estimates. The results showed sharp increases in dust concentration when part of the cleaning plant was closed down.

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NEWS OF THE MONTH

PERSONAL

Mr. Frank Banfield, Mr. Edgar J. A. Flight and **Mr. Harry P. Salmon** have been elected directors of the Expanded Metal Co. Ltd., Burwood House, Caxton Street, London, S.W.1. Mr. Banfield has also been appointed general manager of the company.

Mr. J. W. Barrow, A.M.I.P.E., has been appointed deputy general manager of the M.S.S. Recording Co. Ltd., Colnbrook, Buckinghamshire. Since 1949 until his present appointment, Mr. Barrow has been connected with the development of the overseas factories of British Insulated Callender's Cables Ltd.

Mr. P. J. C. Bovill, Mr. S. C. Tyrrell and **Mr. K. E. Walker** have been appointed assistant managing directors of the chemicals, excavator, and engineering branch, respectively, of Newton, Chambers & Co. Ltd., Thorncliffe, Sheffield.

Mr. Henry D. Challen, A.M.I.Mech.E., director of Taylor & Challen Ltd., has been re-elected chairman, and **Mr. A. E. Whyman,** chairman and managing director of E. W. Bliss (England) Ltd., has been re-elected vice-chairman of the British Power Press Manufacturers' Association, Standbrook House, Old Bond Street, London, W.1.

Mr. A. A. Cross has succeeded **Mr. T. C. Hale** as plant manager of the Harts Lane, Barking, factory of the Cape Asbestos Company. Mr. Hale was recently elected to the Board of the company as Director (Technical), and is now at the company's head office at 114 Park Street, London, W.1.

Mr. L. G. T. Farmer has been appointed assistant managing director, **Mr. Allan Botwood** sales director, **Mr. A. O. Hollick** personal assistant to the sales director, **Mr. G. Lloyd Dixon** general sales manager, **Mr. A. B. Smith** chief buyer, and **Mr. C. Smith** secretary of The Rover Co. Ltd. **Mr. E. G. Commander,** who retired from the position of chief buyer after 58 years' service with the company, has been elected to the Board.

Mr. G. Henderson, A.M.I.E.E., has been appointed general sales manager of Bruce Peebles & Co., Ltd., Edinburgh. **Mr. N. L. Jones** has been appointed London manager of the company.

Mr. W. L. James has been elected chairman of Mellor Bromley & Co. Ltd., Leicester, in succession to the late **Mr. H. R. Backhouse.**

Mr. F. H. Keenlyside has been appointed general manager of the Union Castle Steamship Co. Ltd., 3 Fenchurch Street, London, E.C.3.

Mr. W. T. Kitching, M.I.Mech.E., is now associated with Campbell, Gifford & Morton Ltd., 161 Queen's Road, Weybridge, builders of the first continuous casting plant in Sheffield.

Mr. Leslie Lowe, B.Sc.(Eng.), M.I.C.E., has been appointed chairman and managing director of Davies, Middleton & Davies Ltd., Cardiff.

Mr. B. S. Pelton, B.Sc., A.M.I.E.E., has been appointed manager of the materials handling department of the Fraser & Chalmers Engineering Works of the G.E.C., in succession to **Mr. Oliver Thomas,** who has retired after nearly 52 years of unbroken service with the company.

Mr. Charles Reid, D.S.O., M.A., C.A., has been elected chairman of the Electric Construction Co. Ltd., Bushbury Engineering Works, Wolverhampton, in succession to **Mr. W. M. Furniss,** who retired at the age of 83 after more than 50 years of service with the company. **Mr. W. M. B. Furniss** has been appointed managing director.

Commander (E) A. H. Robinson, R.N.V.R. (Retd.), has been appointed London branch manager of Associated British Oil Engines (Marine) Ltd. **Mr. K. Schwarz, B.Sc.,** has returned to the head office, Ashton-under-Lyne, on the staff of the sales director.

Mr. W. W. Robinson has been appointed sales manager of the Raglan Engineering Co. (Nottingham) Ltd.

Mr. H. O. Sampson has been appointed head of technical operations, television studios, British Broadcasting Corporation.

Mr. C. V. Seymour has been appointed manager of Kerry's (Engineering) Co. Ltd., Leyton, London, E.10.

Mr. Leonard H. Short, M.C., M.I.E.E., M.Inst.T., has been appointed general export manager of The English Electric Co. Ltd., Queens House, Kingsway, London, W.C.2, in succession to **Mr. H. S. Aspinall,** who retired on December 31st. Mr. Short is also a director of the English Electric Export and Trading Company.

Mr. J. E. Sunderland, A.C.A., has been appointed director and general manager, and **Mr. M. J. Smith, O.B.E., B.Sc.(Eng.), A.M.I.E.E.** technical director, of Enfield Cables Ltd.

Mr. T. W. J. Temple has been appointed manager of the Middlesbrough branch office of the British Thomson-Houston Co. Ltd., to succeed **Mr. G. Lloyd Jones,** who has been appointed manager of the BTH Cardiff District Office, succeeding **Mr. A. F. MacCarthy, M.I.E.E.,** who is retiring from the BTH organisation.

Mr. W. G. Thomas, M.Inst.F., has been appointed a director of Radiovisor Parent Ltd., 1 Stanhope Street, London, N.W.1. Mr. Thomas is also managing director of Bailey Meters & Controls Ltd. and chairman of the British Industrial Measuring & Control Apparatus Manufacturers' Association.

Mr. A. E. Underwood, executive director of The Plessey Co. Ltd., Ilford, Essex, has been appointed a director of Plessey International Ltd., the overseas sales organisation for the products of the Plessey Group of Companies.

Mr. H. L. Wickes, A.M.I.E.E., has been appointed sales specialist (motors) of The Brush Electrical Engineering Co. Ltd., Loughborough. He was previously sales manager of Higgs Motors Ltd., Birmingham.

Mr. J. M. Wilson has been appointed works manager of Bowmaker (Plant) Ltd., Willenhall, Staffs.

BUSINESS NOTES

The Federation of British Industries has accepted the invitation of H.M. Government to organise a **British Trade Fair in Iraq.** It will be held in Baghdad from October 25th to November 8th, 1954.

British Overseas Fairs Ltd., will handle the Baghdad and other trade fairs. Chairman of this new company is Mr. Leslie Gamage, M.C., with Lt.-Col. Lord Dudley Gordon, D.S.O., Mr. W. P. N. Edwards, C.B.E., and Mr. P. F. D. Tennant, C.B.E., on the Board. Mr. D. A. Lamb has been appointed head of the organising staff.

Durable Tools Ltd., Avenuepark Street, Glasgow, N.W., is a newly formed company, which will continue the manufacture of collapsible taps and other tools previously manufactured by the Durable Engineering Co. Mr. J. B. Burgess, B.Sc., is managing director of the new company, which has retained the services of Mr. H. R. Macintyre as works manager.

The Brush Electrical Engineering Co. Ltd., Loughborough, have opened a new branch office at 4/5 Lloyds Chambers, Lloyds Avenue, Ipswich, to serve East Anglia. Tel. No.: Ipswich 2749. Mr. G. M. Ingram has been appointed manager for the area.

British Insulated Callender's Construction Company Limited have recently moved into new premises at 30 Leicester Square, London, W.C.2. Their telephone number is TRAfalgar 7777. The central administrative offices of the parent company, British Insulated Callender's Cables Limited, remain at 21 Bloomsbury Street, London, W.C.1.

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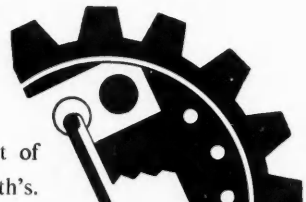

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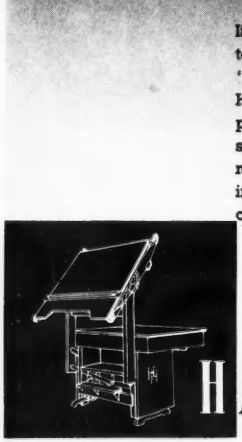

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The Birmingham Welding Co. Ltd. have decided to adopt their long-established registered name of "Birwelco" as the company title, and are now to be known as "Birwelco Limited."

Birfield Industries Limited have further extended their industrial activities by acquiring the whole of the shares of Kent Alloys Limited, of Rochester. The new directorate is Mr. Herbert E. Hill (Chairman), Mr. V. C. Hanson and Mr. C. E. Lovelace (Resident Directors). Additionally, Mr. G. T. Hyslop, Mr. D. L. Roome, and Mr. W. Smithson, of the Phosphor Bronze Company Limited, Birmingham, also a subsidiary of Birfield Industries Limited.

Brooks & Walker Ltd., Engineers' Suppliers, 47 Great Eastern Street, London, E.C.2, announce the removal of their Kidderminster branch to 378 Stourbridge Road, Broadwaters, Kidderminster. The telephone number remains Kidderminster 2016.

MISCELLANEOUS

The Owen Organisation has recently published a Film Catalogue which portrays the large number of films that have been produced for member companies of this Group, which, with its parent company of Rubery, Owen & Co. Ltd. at Darlaston, South Staffs, is composed of nearly forty engineering firms.

In this catalogue are films illustrating the manufacture of parts for the motor industry, all types of agricultural equipment, mechanical handling appliances, and the full story of the B.R.M. racing car.

A colour film is at present in production which will present The Owen Organisation as a whole, and it is hoped that this will be available during the spring.

The Mond Nickel Fellowships Committee now invites applications for Fellowships of an approximate value of £900 to £1,200 for 1954. Fellowships will be awarded to selected candidates of British Nationality with degree or equivalent qualifications, to enable them to obtain wider experience and additional training in industrial establishments, at home or abroad, to make them more suitable for future employment in senior technical and administrative positions in British Metallurgical Industries. Each Fellowship will cover one full working year. Applicants will be required to state details of the programme they wish to carry out. Particulars and forms of application are available from: The Secretary, Mond Nickel Fellowships Committee, 4 Grosvenor Gardens, London, S.W.1.

Completed application forms are required by 1st June, 1954.

Fan Manufacturers' Association Limited. As a non-profit-making organisation, not interested in price rings, the Fan Manufacturers' Association Limited, of 59-62 High Holborn, London, W.C.1, seeks to promote a knowledge of fan engineering, without in any way directing individual manufacturers as to prices.

The Association has produced a new publication, entitled "Handbook of the Fan Engineering Industry" (price 3/6 post free), which, though it does not pretend to refer to all applications for which fan equipment is supplied, is nevertheless sufficiently comprehensive to fulfil its purpose as a link between the industry and the fan equipment using public.

Included among the subjects discussed in this interesting and well-illustrated 58-page publication are the following: Types of fans; air heaters; air cleaners; ducts and fittings; applications for ventilation of industrial and other buildings, of passenger and crew accommodation and cargo spaces in ships, of mines, and of tunnels; industrial dust and refuse removal; drying; land boiler, kiln, and marine boiler draught; forge and cupola blast; air and gas boosting and supercharging; pneumatic conveying; and fans for cooling towers, wind tunnels, and engine testing.

Houseman & Thompson Limited. A maritime and industrial service with such an international coverage as that of Messrs. Houseman & Thompson Limited, requiring a very high standard of scientific and engineering technology, manufacturing and chemical control, and an advisory responsibility to clients, is by no means easy to explain in words.

Their instructional film No. 1, entitled "The Houseman Service," presented in London on January 28th, is an attempt—and, let it be said, a highly successful attempt—to overcome this difficulty by the infinitely more elastic medium of cinematography. The film, which is in colour and runs for about 75 minutes, concentrates the many problems of all industrial water users into one common denominator—"raw" water, and, from the catchment of natural precipitation to its industrial uses, shows many of the effects of untreated or insufficiently treated water in a variety of applications, as widely divergent as steelworks, bakeries, collieries, and hospitals.

Demonstrations of the effects of scale formation and corrosion caused by impure water are strikingly presented in a series of impressive close-ups. On the other hand, the protection afforded, by Houseman & Thompson treatments is also clearly illustrated, not only in the case of boilers and feedwater systems, but in motor car engines and even pithead baths. Chemical descaling and degreasing are other aspects of their service covered in the film, and included in the methods illustrated is the cleaning of modern aero-engines.

The production does not pretend to cover every aspect of water usage and treatment, but it certainly succeeds in promoting a clearer understanding of the scientific approach to water treatment, chemical descaling, and industrial degreasing. In what, for them, must be regarded as a new venture in the field of public relations, Messrs. Houseman & Thompson Limited are to be congratulated on the high standard of this film, though a little judicious cutting, perhaps, would not come amiss. Certainly, any subsequent production sponsored by them on a similar theme will be awaited with considerable interest.

Garringtons Limited. The premiere of the film "They've Come a Long Way" took place on February 3rd in London for the benefit of the Technical Press. This well-produced film traces the development and progress of this long-established engineering firm from its inception in 1840 until the present day. At that time, it consisted of only one small shop, containing a few hammers and machine tools, the bulk of the business being forgings for the tube trade. To-day, the Company is recognised as the largest in Europe producing forgings for the agricultural, motor-transport, automobile, and engineering industries.

Much of the film is devoted to these activities, and the scenes depicting the progress from the costing and estimating stages to the actual tooling-up for mass production, are particularly instructive.

The Fourth Mechanical Handling Exhibition and Convention, organised by the journal "Mechanical Handling", will be held at Olympia, London, from June 9th to 19th, 1954.

Metropolitan-Vickers Electrical Co. Ltd. has recently recieved an order for the electrical equipment of a 40-in. blooming mill at the Cleveland Works of Dorman Long & Co. Ltd.

The equipment comprises two 2500-hp 40/90-rpm motors, one of which is coupled directly to the top mill roll, and the other to the bottom roll; a Ward-Leonard motor-generator set, comprising two 2000-kW generators, a 150,000-hp-sec flywheel, and a 4000-hp slipring induction motor; a main exciter set; and special, quick-response auxiliary exciters for use in conjunction with Ward-Leonard control equipment. The flywheel and induction motor operate in conjunction with an automatic liquid slip regulator in such a way as to reduce the electrical peak loads on the supply system. The main machines are force-ventilated with filtered air, to ensure the exclusion of steelworks dust.

The control scheme has been specially developed for rapid starting and reversal of the main motors, and employs metadyne high-response exciters. Provision is made for automatic load limiting, individual speed adjustment of the motors, and voltage compensation on the generators.

The Mazda Lamp Catalogue this year lists over 1500 of the more important lamps in the standard Mazda range. Full details of each lamp are clearly tabulated, while sections are included dealing briefly with Mazda lighting equipment and fittings, information on other Mazda lamp publications, purchase tax, etc.

Prominent among noteworthy additions to the standard range is the Mazda 75-W reflector spotlight, probably the first light source of its kind to fit an ordinary domestic lampholder. A new range of prefocus flashlight lamps and a lens end flashlight lamp, for use in miniature torches, have been introduced. The range of lamps for vehicles has been modified and increased, as also has the range of decorative lights.

The opening pages and a four-colour centre spread are devoted to the story of BTH lamp production and quality control methods.

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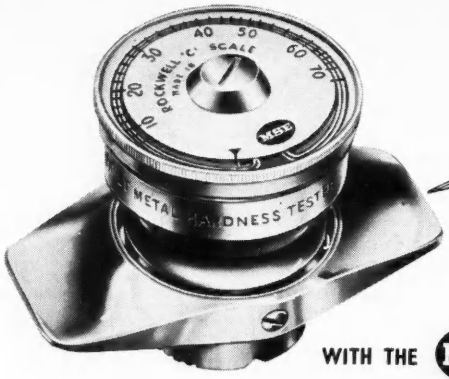

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BOOKS RECEIVED

Battery Chargers and Charging. By Robert A. Harvey, B.Sc.(Eng.), A.M.I.E.E. 400 pp., 284 illustrations. Publishers: Iliffe & Sons Ltd., Dorset House, Stamford Street, London, S.E.1. Price: 35 - (postage 1/2).

Although there has been steady progress in the design of storage batteries during the last twenty-five years, there have been, during the same period, fundamental changes in the methods used for battery charging and control.

This book describes all these new methods, together with the older methods in current use, and shows how battery control problems have been solved in many industries. The construction and chemistry of each type of storage battery are explained, and a description of the fundamental principles of charging is included, together with much general information on charging technique. Specialised applications are discussed, and include electric vehicles and locomotives, emergency lighting and power systems, power stations, telephone exchanges, mines, ships, aircraft, railways, trolley-buses, cars and commercial vehicles, and generating plant for country houses and other isolated buildings. Home chargers for car and radio batteries, and charging arrangements for commercial garages and radio and electrical ships, are dealt with in a special chapter. Reference and design data are given in appendices.

This book will be of interest to all concerned with the installation, operation, and maintenance of battery charging systems in any field, and particularly to engineers considering new or unfamiliar applications.

Introduction to Ultra-High-Frequency Radio Engineering. By Stephen A. Knight, F.R.S.A. 264 pp., 202 illustrations. Publishers: Sir Isaac Pitman & Sons, Ltd., Parker Street, Kingsway, London, W.C.2. Price: 21/-.

Radio engineering has, in the past decade, changed profoundly in the nature and complexity of the apparatus and techniques upon which it was originally founded. Many of these have developed under the pressure of military requirements, but the field of application to civil and commercial projects is still comparatively unknown to the general engineer and the radio amateur.

This book is intended as a guide to the methods and techniques of modern ultra-high-frequency radio and radar engineering, and is written primarily for those engineers, students, and members of the technical branches of the Services who wish to obtain a working background of the subject, without complicated mathematical theory or unnecessary detail. As such, it will fill the gap between books aiming to teach the subject to students with little or no grounding in radio and electricity, and those written for the practising research engineer. All that is assumed is a knowledge of elementary radio systems and general techniques.

The author, who has had many years of experience in the practical side of radio engineering, covers such typical aspects as transmission lines and waveguides, resonant cavities, cathode-ray tubes, disc-sealed valves, klystrons and magnetrons, measurements, aerial systems, and wave propagation.

Industrial Training for Manual Operations. By W. Douglas Seymour. 214 pp., 30 illustrations. Publishers: Sir Isaac Pitman & Sons, Ltd., Parker Street, Kingsway, London, W.C.2. Price: 20/-.

This book gives an account of some recent developments in industrial operator training, and provides an outline of the basic factors affecting industrial skill and its acquisition. The aim of the book is to provide the manager or supervisor in industry with an outline of the mental and physical processes involved in such acquisition, and also to serve as a guide on how to set about training workers on repetitive operations.

In addition to many years of successful practice as an industrial consultant, specialising in personnel and training problems in many different types of industry, the author has conducted active research work on the nature and acquisition of industrial skill, as Tube Investments Research Fellow in the Department of Engineering Production at the University of Birmingham.

A Manual of Time and Motion Study. Fourth edition, 1954. By John W. Hendry, F.R.Econ.S., F.C.I.M., M.I.W.M. 231 pp., 48 illustrations. Publishers: Sir Isaac Pitman & Sons, Ltd., Parker Street, Kingsway, London, W.C.2. Price: 15/-.

All branches of industry require standards of measurements peculiar to their particular needs, and it is safe to say that at no other time has a standard of measurement been needed as much as it is nowadays. Accuracy is the keynote of production and, without it, the complex industrial machine of to-day would never have grown to such formidable proportions.

Time study as a measurement standard, and motion economy as an aid to the efficient use of labour, are two subjects which are growing more and more in importance and are beginning to pay handsome dividends to those who employ them honestly, fairly, and with a keen understanding of human relations.

This book should prove valuable in contributing to raise efficiency standards in industry and should find a place on the shelves of all industrial managers.

Entwicklung der Räder für Lastwagen und Omnibusse. Compiled by Dipl. Ing. H. Buss. 188 pp., 240 illustrations. Published by George Fischer Limited, Schaffhausen, Switzerland.

One hundred and fifty years ago, Johann Conrad Fischer founded the George Fischer Works, and fifty years ago the firm commenced manufacture of cast-steel spoked wheels for motor lorries. This book (in German) on the development of wheels for lorries and omnibuses has been issued to mark this dual Jubilee, and recounts the story of the development of wheels in general, and Fischer wheels in particular.

The oldest representations of wheels, which L. C. Woolley, the British archaeologist, discovered in 1929 in the deeply buried tombs of the kings in Ur, the old capital of the Sumerian kingdom of the Euphrates, already reveal a design conceived of compound structure. From that time onwards, the spoked wheel has always proved superior to the primitive solid wheel. An outstanding example is the bronze wheel of Cortaillod, the most ancient wheel discovered in Switzerland, and a forerunner of modern cast wheels.

The first part of the book covers, therefore, more than five thousand years history of the wheel; the second part is concerned with Fischer cast wheels for solid rubber tyre equipment, manufactured after the turn of the century for motor lorries, while the third part is devoted to wheels equipped with pneumatic tyres.

The book is magnificently produced and illustrated, and not the least fascinating thing about it is the wealth of superb illustrations of ancient wheels and of wheels and vehicles in the "primitive" era of the internal combustion engine.

Protective Atmospheres. By A. G. Hotchkiss and H. M. Webber. 351 pp., 167 illustrations. Publishers: John Wiley & Sons, Inc., New York, and Chapman & Hall, Limited, 37 Essex Street, London, W.C.2. Price: 56/-.

Protective atmospheres can save money, improve quality, and speed production in industry. For example, furnaces, with protective atmospheres, produce work free from oxides or scale, thereby saving metal and eliminating cleaning operations, while in the paint and varnish industry, such atmospheres prevent oxidation in storage and eliminate skins on paint and varnish in tanks and pipelines.

This book has been written to assist those who supervise manufacturing operations to obtain a usable understanding of the various phases of protective atmospheres; as such, it should be a valuable aid in helping to select the best atmosphere gases for specific requirements and to apply and use them wisely and safely, so as to produce the results desired. The book is not intended to be a scientific treatise on theory or on equipment design; indeed, emphasis has been placed, rather, on practical aspects, so that it can serve as an operating manual.

The ability to diagnose troubles and to check them quickly is important to all users of protective atmospheres. Simple tests to determine the causes and results of discolouration, oxidation, or decarburisation are invaluable for finding and eliminating operational faults. This subject of protective-atmosphere ailments and their remedies is well covered in the book. Also included are data, tabulated for ready reference, showing compositions and costs of typical protective-atmosphere gases, common industrial processes for which they are used, materials processed, requirements met, and recommended atmospheres for specific applications.

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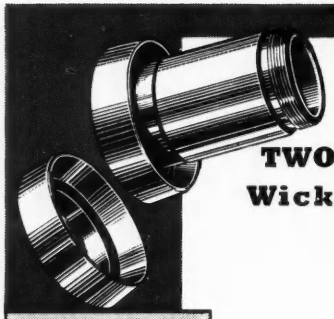
ERRATUM

The Cycle Controller described in the "New Materials, Processes and Equipment" Section of our issue for January, 1954, was inadvertently stated to be manufactured by the Bristol Instrument Company, of Bristol, instead of by Bristol's Instrument Company Limited, of Weymouth, Dorset.



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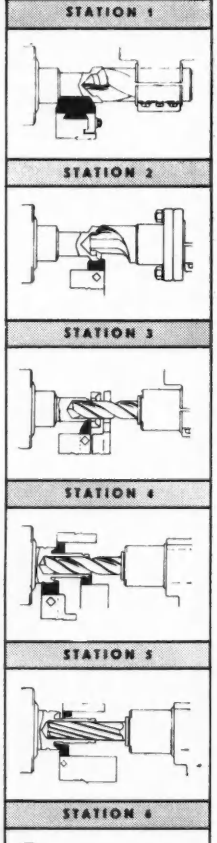
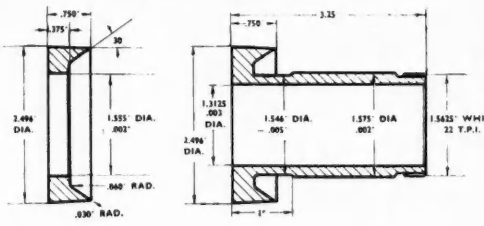


Commutator Hub and Clamping Ring (Mild Steel).



So often the production engineer thinks in terms of one part, and one machine to make it . . . and although it may be the exception rather than the rule to make two parts together on a Wickman Automatic, this example demonstrates the outlook of Wickman Engineers in planning for maximum machine utilisation, and the exceptional tooling opportunities offered by the range of Multi-spindle Automatics.

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BRITISH STANDARDS

(Copies of British Standards may be obtained from the British Standards Institution, 2, Park Street, London, W.1.)

Gear Hobs. B.S. 2062 : 1953 covers in Part 1 the range 1 to 50 D.P. for general purpose hobs intended for the production of gears complying with the basic rack tooth profile given as Fig. 5 of B.S. 436 : 1940. Three grades of accuracy, i.e., precision-ground hobs, ground hobs and unground hobs, and the tolerances in Table 4 are in close agreement with those of the American Gear Manufacturers' Association.

Appendix B lists the sizes of hobs with taper bores and of taper arbors in current use.

Part 2, with Table 5, sets out limits of *accuracy only* for hobs intended for the production of gears complying with B.S. 1807 : 1952 "Gears for turbines and similar drives, Part 1 : Accuracy." This table is based on the specification issued by the Mechanical Engineering Research Laboratory of the Department of Scientific and Industrial Research for the "Accuracy of gear cutting hobs for marine turbine and similar drives" and which has been in use for several years.

Part 2 does *not* cover tooth form design. Price 5/-.

Copper and copper alloy capillary and compression tube fittings for engineering purposes. B.S. 2051, Pt. 1 : 1953 relates to three types of tube fittings for engineering purposes of sizes ranging from $\frac{1}{8}$ in. to 1 in. inclusive, for use in conjunction with copper, copper alloy and other suitable tubing, *viz.* :

a. Capillary fittings in which the joint is made by the flow of solder by capillarity along the annular space between the outside of the tube and the inside of the socket of the fitting, the size of this annular space being dimensionally accurate within close limits.

b. Compression fittings, Type "A," in which the joint is made by the compression of a ring or sleeve or part of the fitting, on to the outside wall of the tube.

c. Compression fittings, Type "B," in which the joint is made by the compression of a manipulated portion of the tube at or near its end against true faces of the body of the fitting or against a loose ring or sleeve within the fitting.

This standard does not attempt the complete dimensional standardisation of any of the types of fittings, since the variety in the designs and methods of production already established by the various manufacturers makes any such attempt impracticable. Without seeking to limit the individual refinements of design, which are the distinctive feature of the proprietary product, this standard does, however, lay down such dimensions and requirements as are essential to ensure satisfactory installation and performance.

The fittings covered by this standard are for use with tubes designated by their outside diameters in inches and fractions of an inch. Price 3/-.

Balata Belting. B.S. 2066 : 1953 specifies requirements for balata belting used for power transmission purposes and other load bearing applications, e.g., straps and slings.

This standard also specifies details of the fabric, impregnation, balata content, quality of balata, etc. and includes appendices giving details of methods for testing. Additionally, appendices are included concerning working conditions and information to be given with the enquiry or order. Price 2/6.

Guide to the selection of fits. B.S. 1916, Part 2 : 1953 provides reliable data in the form of recommended tolerances which can be used for a wide variety of standard applications.

The introduction of a new limits and fits system is inevitably a long-term project and many users, faced with the wide choice of fits in Part 1, might feel that a long period of experiment was necessary before they could decide which tolerances to adopt as standard. Reference to Part 2, with its selection of clearance, transition and interference fits for many typical applications, should go far to solve this problem as well as minimising the possibility of differing tolerances and allowances being chosen to meet the same design problem.

The guide is fully illustrated and includes much useful information regarding the various grades of fit, the accuracy which can reasonably be expected from various manufacturing processes, etc. It also contains tables of preferred fractional and decimal inch sizes, the use of which is indispensable to true standardisation.

The recommendations given in the guide are based on the results of investigations carried out with the co-operation of a number of engineering firms and organisations. It is recognised that some requirements are not yet adequately covered, and an early revision of the guide to include more material is envisaged.

In order that it may be as practical and helpful as possible, users are earnestly requested to submit to the Director of the Institution their comments, criticisms and constructive suggestions for consideration by the responsible committee. Any information supplied will be treated confidentially and will be of great assistance in rendering this guide of real value to industry.

Straight-sided splines and serrations. B.S. 2059 : 1953 presents the dimensions of straight sided, bottom fitting, shallow and deep splines, together with those for straight-sided 90° coarse serrations.

In both cases the figures in the tables allow a choice of three grades of fit and are followed by provisions for Full Form Go gauges and Not Go gauges for the individual features.

The Standard is divided into two Parts, the first for splines and the second for serrations, each part being completed by several appendices, wherein will be found much useful explanatory matter, fully illustrated. Price 12/6.

Determination of Power Factor and Permittivity of Insulating Materials. B.S. 2067 : 1953 describes in detail the theory and technique of the Hartshorn & Ward method for the determination of power-factor and permittivity of insulating materials at radio frequencies.

This method has been adopted as a standard test in certain British Standards for insulating materials, and it is suitable for the measurement of power-factor and permittivity within a frequency range of 10 kc/s to 100 Mc/s at normal room temperatures. The method is essentially one of capacitance variation in a tuned circuit with a thermionic voltmeter as a detector of resonance. Details are given of the technique to be followed in testing moulded and sheet material, tubes and liquids, and drawings are included to illustrate the basic test circuit and some of the basic equipment involved. Appendices describe the theory of the method and detail the essential features of the main components forming the test assembly. Price 4/-.

Classified Advertisements.

The rate for all classified advertisements is 6d. per word; in bold print 9d. per word; minimum order 6s. Box number advertisements 1s. extra. Instructions together with remittance must be received not later than the 3rd of each month for advertisements to appear in the same month's issue.

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WANTED.—Copies of the March, 1953 issue of "The Engineers' Digest" urgently required. Also Jan., 1949, June, 1950, and Dec., 1951, issues at 3/6 per copy.—Subscription Dept., 120, Wigmore Street, London, W.1.

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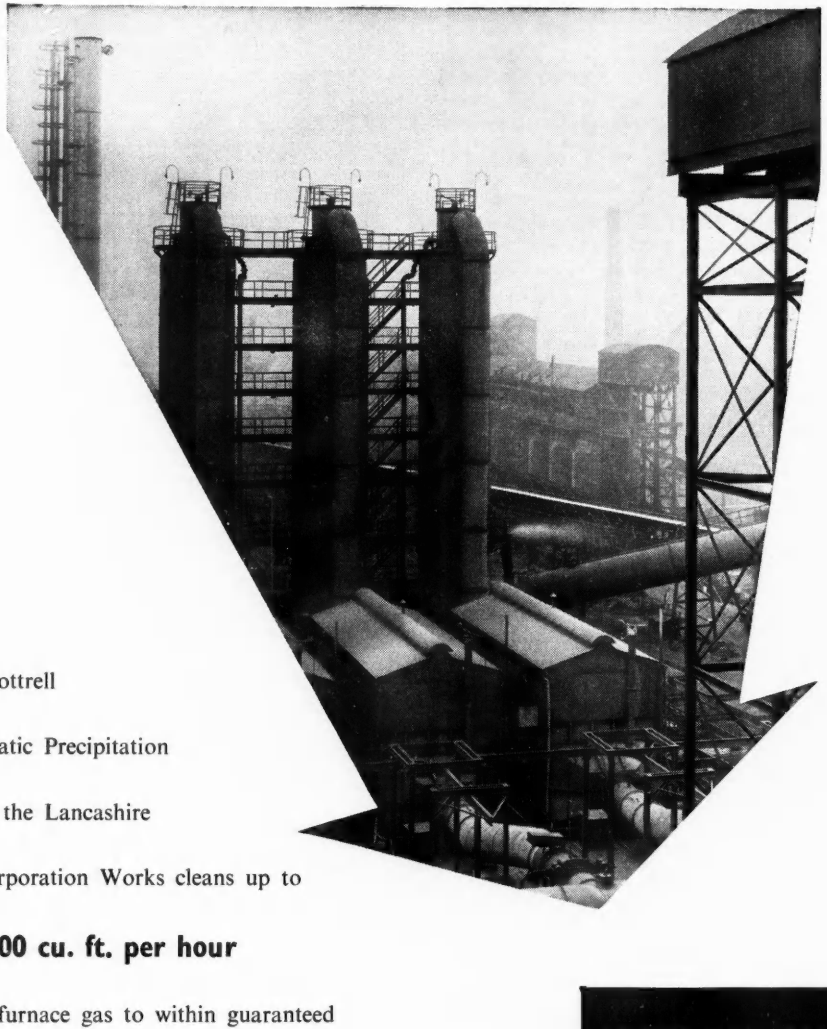

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LATEST INDUSTRIAL LITERATURE

1. Melting and Heat-Treatment Plant. The first of a new series of catalogues and brochures on melting and heat-treatment plant is now available. These publications will prove useful for reference purposes, since, besides details of equipment, considerable metallurgical and technical information is included. A folder, describing recently introduced air-conditioning equipment for continuous operation in self-contained areas, is also available, together with a pocket-sized folder of humidity tables.

The series includes revised brochures dealing with shaker hearth conveyor furnaces; gas carburising; electric furnace brazing; and the "Machine tool approach to surface hardening." In addition, there is a new 20-page descriptive catalogue of arc furnaces and a publication dealing with induction heating equipment.

2. Twist Drill Fluting Machines. A leaflet describes a range of twist drill fluting machines with multiple heads, and triple spindle twist drill fluting machines.

The machines with multiple heads are supplied as single units, or with one, two, or three heads mounted on one bed. They represent the simplest form of drill fluting machine, and can be set up and operated by unskilled labour. The triple spindle machines flute three drills at the same time, and extra chucks are provided for loading while the machine is in operation.

In both cases, $\frac{1}{2}$ in. and $\frac{1}{4}$ in. models are available.

3. Crucible Furnaces. A 4-page brochure describes and illustrates a range of crucible furnaces, fired by oil, gas, or coke. The range includes hydraulically tilted, lip pour furnaces; lift-out furnaces; rotary crucible furnaces; miniature furnaces; central axis tilting furnaces; bale-out furnaces; and crucible heaters.

With this range of furnaces, special supervision and skilled melting techniques are not required on regular production in normal circumstances. Furthermore, the melting is carried out under metallurgically clean conditions.

4. Fans and Heating Units. A range of fans and unit air heaters is described and illustrated, together with performance data, in a neat 8-page brochure. The fans are of the centrifugal type and are available for all kinds of applications. The range of air heaters includes battery or coil heaters, which can be used in service as preheaters, reheaters, main heaters, and booster heaters, and whose function is largely associated with air conditioning, drying, ventilating, etc.

In addition, a radiator is available, suitable for steam heating circuits up to 100 psi, and with outputs from 3600 to 26,300 B.Th.U./hr.

5. Metal Shelving. A folder describes and illustrates a range of metal shelving, a feature of which is easy and speedy erection, and rapid shelf adjustment without the use of nuts and bolts. No less than 75 standard sizes are available, so as to provide increased storage capacity, as required.

A range of bolted type open and closed shelving is also described. This shelving is available in standard widths of 2 ft to 3 ft 9 in., and standard heights of 3 ft 3 in. to 10 ft 3 in. In addition, a range of tool and clothes lockers as described and illustrated.

6. Plastic Mouldings and Zinc Alloy Die Castings. In a neatly presented and illustrated 16-page publication, the fundamental advantages and disadvantages of plastic moulding, as they exist to-day, in comparison with zinc alloy die castings, are detailed. The use of zinc alloy die castings in conjunction with moulded plastics, both as relatively small inserts and as castings representing a large proportion of the composite product, is also discussed.

A supplementary series of publications deals with zinc alloy castings and productivity, with the commercial applications of zinc alloy gears, and with the welding of zinc alloy die castings, respectively.

7. Fluid Valves. The choice of fluid valves in the past, for installation in pipe lines ranging from 6 to 36 in. bores, has been largely confined to general types, *viz.*: Sluice or gate valve, and barrel valve, respectively. A new valve, specifically intended to cope with present-day requirements, is described in a four-page leaflet. This valve, manufactured in ranges between 9 in. and 36 in., is suitable for oil, air, water, steam, etc., and for high-pressure and low-pressure applications.

The whole design is symmetrical, ensuring that expansion and contraction under heat variations are uniform, both in respect of valve and seating. Its compactness is exemplified by the fact that a standard 36 in.-bore valve is only 9 in. in width.

8. Wear- and Corrosion-Resistant Deposits. Details of a process for the electrolytic deposition of chromium are contained in a well-produced and illustrated 20-page booklet. This process has been developed to ensure that the properties of chromium, when electrolytically deposited under carefully controlled conditions, are consistently applied to industrial requirements. These properties include hardness, low coefficient of friction, and high resistance to corrosion and abrasion; also, as no heating is involved in the process, freedom of distortion is assured.

Furthermore, as deposition is made directly on the base metal, a perfect bond between base metal and deposit is assured. The deposit can be applied locally, so that only those parts requiring deposition are treated.

9. Paint Transfers. A range of transfers for practically every conceivable use in industry and commerce is described in a well-produced 12-page booklet. In addition to industrial transfers, the range includes decorative transfers for the home, the nursery, etc., and institutional transfers for such purposes as crests, regimental bades, and insignia.

These transfers are easy to apply on any smooth surface; they are washable and heat-resisting, and will withstand the most severe conditions of weather and usage.

The booklet is profusely illustrated in colour with representative names and trade-marks of well-known firms and products, reproduced by these transfers.

10. Red Lead Pigments for Paints. To provide the maximum protection for iron and steel, an anti-corrosive primer should include one of the limited number of corrosion-inhibiting pigments available and should retain good adhesion throughout the life of the paint film; furthermore, the film must maintain its flexibility and should be tough and continuous. Red lead, used either as the sole pigment or in combination with other pigments, provides a paint film which fulfils all the above requirements.

A large variety of red leads is described in an excellently produced and illustrated 36-page publication. In addition to listing the properties of red lead in anti-corrosive paints, there are interesting and instructive chapters dealing with the composition and manufacture of red lead, types of red lead, formulae and methods of preparation of red lead paints, red lead oil vehicle primers, and quick-drying red lead paints.

A similar publication of 24 pages deals with the use of red lead paint against corrosion.

11. Resistance Welding. The most careful adjustment of current and pressure settings on a welding machine can quickly be negatived if its electrodes become badly worn or deformed. The combination of specialised materials with advanced electrode design ensures that users obtain the best from their welding equipment.

A comprehensive range of electrode and die materials is described in a 28-page booklet. In addition, the booklet deals in turn with the four main welding processes, i.e., spot, seam, projection, and flash and butt welding, in terms of the most suitable choice of material, the design and construction of the electrodes, and the recommended practice for obtaining maximum efficiency from machine, material, and electrode.

12. Application of CO_2 to Machining Operations. The suggested application of carbon dioxide to machining operations is not new, but up to now it has not been entirely satisfactory in practice. However, in a well-produced and illustrated 16-page booklet, a new technique is described, which marks a considerable advance in this field, and particularly merits the attention of those concerned with the working of modern high-strength materials.

Although CO_2 cannot take the place of the more familiar liquid coolants in some instances, there is nevertheless an extremely wide field for its development. Some of these aspects are discussed in this publication, particularly with relation to its use in turning, drilling, milling, boring, and tool tip grinding operations.

13. Controlled Electric Clock Systems. Full details are given in a 36-page booklet of a controlled electric clock system, in which an accurate, precision master-clock is used for controlling any number of "slave" clocks and other subsidiary equipment.

This subsidiary equipment includes a comprehensive range of attractively designed wall clocks, of surface or insertion type, and suspension clocks. In addition, there is a range of time recorders, sound signals, watchman's tell-tale clocks, and programme instruments, the latter being used for the automatic operation of hooters, buzzers, bells, etc. at intervals throughout 24 hours.

The source of current in this system can be either high-efficiency dry cells or trickle-charged accumulators, in conjunction with a simple two-wire circuit.

14. Treatment of Hard Water. Details of a process for treating hard water, so as to prevent the formation of scale, are contained in an interesting 24-page, illustrated publication. The equipment used in this process is extremely compact and is made in a large range of standard sizes. It is of simple design, consisting of a pipe unit which is inserted into the pipe line through which all water passes. A control box, adjacent to the pipe unit, controls the electrical feed in accordance with the special requirements of the process, and is energised from a.c. mains or by its own self-contained dry battery, as desired.

Outlay for chemicals can be eliminated, as the equipment has a negligible current consumption (a few shillings yearly); furthermore, the labour required for chemical treatments is unnecessary, and there is no heavy and continuous maintenance expense. The device is automatic in operation, has no moving parts, and is not affected by any variations in load.

15. Angle Plates. Two leaflets detail and describe a range of box angle plates and swivel angle plates, respectively. These angle plates are of great value to toolmakers, gauge makers, etc. and on small surface and gap-grinding machines.

Both types of angle plates are manufactured from high grade, seasoned castings, and are carefully and accurately machined.

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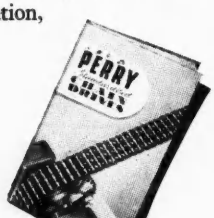

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● *NEW HEAT-RESISTANT ALLOY*

A new casting alloy for operating temperatures as high as 2200° F is claimed to have exhibited excellent service characteristics in a number of trial applications, such as radiant tubes, retorts, muffles, and similar parts. It is well known that many alloys which retain considerable strength at high temperatures lack oxidation resistance, while other alloys with high oxidation resistance lack high-temperature strength. The problem therefore consisted in strengthening a nickel-chromium-iron base alloy and imparting to it increased oxidation resistance. The alloy finally evolved contains nominally 0·5% carbon, 48% nickel, 28% chromium, and 5% tungsten. This alloy is claimed to possess excellent welding characteristics. For welding, cast rods of the same composition are coated with a special low-hydrogen coating and welded with direct current of reversed polarity. No preheating for welding is necessary in the case of thin sections, but for heavier sections a low heat input and frequent stress relieving by peening are advisable. Tests are claimed to have shown the welds to be equal to the cast alloy in high-temperature properties. Radiant tubes for annealing furnaces are reported to have operated successfully at 2000° F for seven years. In the case of endothermic gas generators, the new alloy is making it possible to operate these generators at temperatures of 2050° to 2100° F, whereas previously a temperature of 1850° F could not be exceeded. As a result, the gas output is increased by some 60%, and at the same time the gas is of more uniform composition. Other promising applications are claimed to be the use of the alloy for electric-heater supports in brazing furnaces, for baffle-support bars in steam superheaters, and for soaking-pit burners.

● *MINIATURE ELECTROSTATIC GENERATOR*

Many instruments for the detection and measurement of nuclear radiation require high-voltage sources of low current to supply polarisation potentials for ionisation chambers. Conventionally, this voltage is obtained from miniature high-voltage batteries used directly or in combination with capacitors charged in parallel and then connected in series, from a vacuum-valve oscillator in combination with a high-ratio transformer and rectifier, or from a vibrator power supply. A miniature electrostatic generator has now been introduced which is simpler and less expensive to manufacture and which avoids the use of special batteries. This generator is an adaptation of the well-known Holtz and Wommelsdorf influence-type generators of conventional size, modernised by the use of printed-circuit and miniaturisation techniques. The machine consists basically of a stator of two field plate conductors, and a rotor with a number of pairs of conducting sectors. The conducting areas on the flat insulating plates of rotor and stator are produced by printed-circuit technique. Several sets of brushes transfer electrical charges between the components of the system and the storage capacitor. A driving system of the reciprocating type is used to drive the rotor at speeds as high as 6000 rpm. The reciprocating type was chosen for the ease with which it can be adapted to a hermetically enclosed generator, without requiring sealing glands. A small external bias voltage is used to precharge the generator. It is reported that size places no particular limitation on the successful performance of the printed-circuit type generator, and units as small as $1\frac{5}{8}$ in. in diameter have been built.

● *DETERMINATION OF GRAIN SIZE BY ULTRASONICS*

The last few years have witnessed a steady increse in the application of ultrasonic technique to the non-destructive testing of materials. This technique can also be used to determine the grain size of a metal, as it has been found that the grain size has a marked effect on the transmission qualities. Research investigations revealed that, as the ratio of the ultrasonic wave-length to the grain size was decreased below a certain value, the amount of ultrasonic energy transmitted diminished. In these investigations the range of wave-length to grain-size ratios covered was from approximately 100 : 1 to 5 : 1. At wave-length to grain-size ratios above 30 : 1, the transmissibility remained essentially unaffected by variations in grain size, but below a 10 : 1 ratio the transmissibility rapidly diminished in intensity. No difference was detected between the pulse-echo technique or continuous transmission methods. However, the multiple back-reflection technique proved to be the most useful from a practical standpoint. Impurities or internal defects which influence the amplitude of the reflections do not

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★*At a firefighting demonstration in Glasgow* ★★★★★★★★★★★★

Pye Industrial television was recently used as a method of control in a full scale fire-fighting exercise of the Glasgow Fire Brigade. TV Cameras at the scene of the fire relayed pictures to the Central Headquarters, giving immediate and accurate visual information of the size, condition and progress of the fire.

Observers were enthusiastic regarding the value of television as a new aid to fire fighting. It is not envisaged that local operations would be directly controlled from headquarters, but television removes the difficulties of assessing from verbal reports the size and menace of a particular fire.

This application of television as an aid to Fire-Fighting is only one of the many uses for industrial television. For further information will you please write to Pye Limited, Industrial Television Division, Cambridge.

In basic form the equipment comprises a lightweight television camera and a viewing monitor; these are linked together by a single co-axial cable which may be any convenient length from a few feet to a thousand feet, depending upon the user's requirements.

The camera, weighing only 31 lbs. and measuring 16 x 6 x 14 inches approximately, carries three lenses giving a wide choice of viewing angles and features a newly developed low-cost camera "pick-up" tube possessing adequate sensitivity to normal industrial lighting. In practical terms, a clearly defined picture will be obtained with a scene brightness as low as 50 foot-lamberts—corresponding to the light reflected from an "average" subject placed 4 feet from a 200 watt lamp in a suitable reflector.

The viewing monitor displays a well-defined picture on a 14 inch diagonal rectangular cathode-ray tube with a built-in "black screen" to permit clear viewing with normal room lighting. A normal domestic television receiver may, without modification, be employed as an additional monitor if required.

Alternative versions of the camera and monitor provide scanning standards of either 405 lines or 525/625 lines and the equipment will operate on any normal AC supply voltage, with a total power consumption of only 350 Watts.

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affect the number of multiple reverberations, and the multiple-reflection method therefore appeared more reproducible and less sensitive to test conditions than any other ultrasonic method. A compressed specimen, which had internal residual stresses due to a ten per cent axial deformation, exhibited the same ultrasonic properties as the specimen with the same grain size but no axial compression. Ultrasonic transmissibility effects may thus be attributed to variations in grain size and not to some factor such as the level of stress relief produced by heat treatment. This method makes it possible to ascertain the structural condition of parts exposed to elevated temperatures in service, so that they may be kept in service as long as their grain size proves satisfactory.

● *AUTOMOBILE AIR CONDITIONING*

A new automobile air-conditioning system of advanced design has been placed on the market and is claimed to be unique in several respects. All components of the system are arranged forward of the instrument panel. Only fresh outside air is cooled and delivered to the passenger compartment, and there is no re-circulation. The entire system is fully automatic in operation. A temperature control lever regulates the air supply, connects and disconnects the Freon-12 compressor through an electric clutch, and modifies the temperature, as required; all these actions are carried out in one operation. The cooled air enters the passenger compartment through three outlets on the panel board, these being positioned to suit the convenience and the comfort of the passengers. An independent switch controls the volume of air and is graduated in steps from minimum to maximum cooling, as desired. The condenser, which is of all-brazed construction, is located in front of the radiator and is designed to combine maximum efficiency with minimum effect on the engine cooling system. A small two-cylinder reciprocating compressor is used, its crankshaft being mounted on ball bearings for high-speed operation. The compressor is equipped with a magnetic clutch and is in operation only when actual cooling is required. The evaporator is of the vertical all-aluminium type. By means of a special valve, sufficient refrigerant is maintained in the evaporator to ensure that cooling is immediately available when required. When the system is operating at low capacity, a return refrigerant line and valve are used to provide refrigerant and lubricant to the compressor.

● *IMPROVED MAGNETIC POWDER CLUTCH*

A magnetic powder clutch may be defined as a controllable coupling which utilises the action of magnetic metallic particles in a magnetic field between a driving element and a driven element, both made in whole or in part of magnetic material. The magnetic field produced by passing an electric current through a coil generates a binding force or torque which is approximately proportional to the coil current up to the point of saturation of the magnetic circuit. There are two general types of magnetic powder clutch in use. In the one type, a liquid vehicle, such as oil, suspends the iron particles and serves as a source of lubrication. The oil also acts as a protective coating which inhibits oxidation of the iron powder. This type of clutch suffers from the disadvantage of requiring special seals, the possibility of oil leakage, and sedimentation of the suspended particles. In the other type of clutch, no liquid is used and the iron particles are dispersed in the air gap. Additions, in the form of powders, generally serve as lubricants and prevent physical and chemical changes in the iron particles. According to a recent report, replacing conventional iron powder mixtures by magnetic stainless steel powder, without additives, promises to remove many of the shortcomings now associated with magnetic powder clutches. Previous difficulties, due to erratic performance under the influence of oxidation and contamination with oil and grease from bearings, should be effectively eliminated. Clutches charged with stainless steel powder are reported to have withstood severe humidity tests during prolonged operation, while clutches fitted with ordinary iron powder failed completely after operating for only a few days.

● *ANTI-CORROSION COATING OF SURFACES WITH NICKEL-PHOSPHORUS ALLOY*

It is claimed that a new type of corrosion-resistant coating can be applied to ferrous and non-ferrous metal surfaces of any shape, and the point is made by the inventor that it can even be applied to the interior surfaces of long pipes, heaters, and other bodies which have relatively small openings. The wide range of possible applications of the new coating may be gauged from the claims of the inventor to the effect that coated tubes of this type may be used in oil stills, or that the improved coating may be applied to resistors, grid leaks, emission coatings in vacuum valves, etc. Provided that the surface to be coated is absolutely clean and free from rust, grease, scale, oil, paint, etc., actual interalloying of the coating material with the base material takes place. A further advantage of the new coating would appear to be its ductility, which makes it possible to bend a thin coated sheet without destroying the coating. The major ingredient of the coating is either nickel, cobalt, or copper. A typical mixture is made of nickelous oxide and ammonium di-hydrogen orthophosphate in the ratio of 2·5 to 1, 4 to 1, or 8 to 1. This mixture is made with water or other liquid, sufficient to produce a paint-like consistency, so that, when the mixture is brushed on the surface, a fairly thick coating

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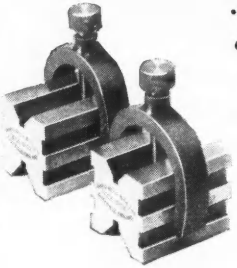

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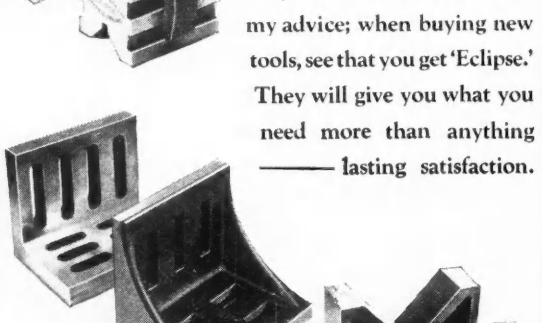

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will remain. The moist coating is then heated in a suitable reducing atmosphere to 1600-1800° F. The action of the heated reducing agent, which may, for example, be pure hydrogen, is to decompose the phosphate and the phosphorus alloys with the nickel. An alloy coating of nickel and phosphorus is then produced, which to some extent is interalloyed with the base material.

● *TIN-ZINC PLATED COATINGS*

In the light of present stages of development, tin must certainly be considered to be the most versatile alloying element available for plating processes. About the only alloy plating processes not involving tin are brass, nickel-cobalt, and some of the gold alloys. On the other hand, alloys of tin with copper, zinc, nickel, lead, and cadmium have been plated commercially, and several others are in the laboratory stage. Tin and zinc are far apart in electrode potential but, by special measures, alloys of tin and zinc of any desired composition can be deposited. Extensive tests have shown, however, that for most purposes the optimum range of composition is a 75 to 80% tin, balance zinc, alloy. Tin-zinc alloys are useful as coatings because of a combination of properties rather than of any single characteristic. Their corrosion resistance is in a class with zinc and cadmium, while their soldering qualities are equal or superior to those of tin. Tin coatings, especially electroplated ones, have occasionally been criticised for lack of good soldering qualities after storage or ageing. In contrast, tin-zinc alloys have been found to be easily solderable after as long as two years storage. The properties of this alloy prove particularly favourable in cases where dissimilar metals are in contact. Thus, for instance, tin-zinc plating of the steel bolts used in aluminium structures largely eliminates the corrosion failures previously experienced. The process for plating this alloy is relatively simple, and the bath used is one combining good efficiency with excellent throwing power.

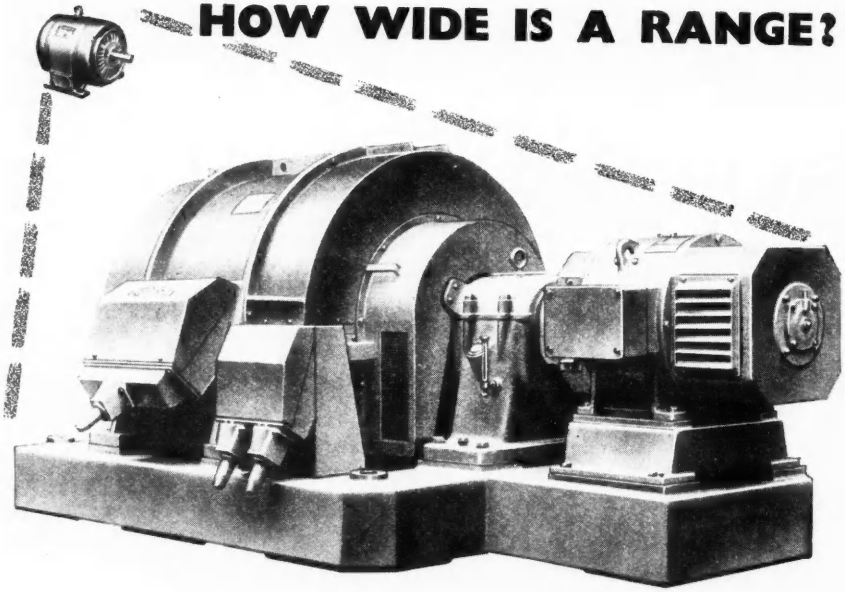
● *HIGH-PURITY CHROMIUM*

There is no doubt that, in the development of new alloys for the steadily increasing operating temperatures of high-temperature prime movers, chromium will retain its important position as an alloying constituent. Despite the wide application of chromium in present-day alloys, the nature and behaviour of this material are not yet sufficiently known. One of the most important problems in this respect is the fact that chromium may have high strength at high temperatures and may be as brittle as glass at room temperature. Latest research indicates that room-temperature brittleness may be due solely to impurities and that, in fact, all records with room-temperature brittleness of this material were obtained on material of a quality far inferior to that of the high-purity chromium obtainable to-day. Commercial chromium may be expected to contain measurable amounts of iron, copper, silicon, and other metallic impurities, plus unspecified amounts of oxygen and hydrogen. In this connection, it is interesting to note that a low-oxygen alloy of 60% chromium and 40% iron was found to be readily forgeable and to have excellent ductility at a tensile strength of 140,000 psi. Production of high-purity chromium has been attempted by several methods. From the aspect of metallic contamination, the purest chromium is that produced by electrolytic deposition. Commercial electrolytic chromium contains 0·5 to 1% oxygen, which can be removed by hydrogen treatment at high temperature. Chromium produced by the electrolytical method was drawn into wire, but was found to lack ductility. However, by etching the wire, a remarkable increase in ductility was obtained, and an investigation of the underlying cause of this improvement in ductility is now in progress.

● *LOW-FREQUENCY INDUCTION HEATING*

Recent improvements in the technique of low-frequency induction heating are opening up new fields for the application of this heating method in the metal-working industry. That applications are not limited to small-size components is shown by the fact that with 60-cps current, steel billets, 13 in. square and 30 in. long, have been heated to a rolling temperature of about 2300° F in less than one hour. The coils used to heat the steel are made of copper tubing, through which water is circulated to keep the units cool. Steel billet heaters have also been designed which utilise two-stage heating, both in single and dual frequency. This design has been introduced in view of the fact that the electrical heating characteristics of carbon steel change radically at the Curie point, above which the material becomes non-magnetic. It is therefore desirable to use separate coils for heating up to and beyond that point. Depending upon billet size, the final heating coil may be designed for any frequency from 50 to 10,000 cps. In general, the larger the diameter of the piece to be heated, the lower is the required frequency. Low-frequency induction heaters are now also used to heat aluminium and magnesium billets up to 32 in. in diameter and over 80 in. long. Most of these heaters are of the vertical-coil type and a heater of this type will heat a 6000 lb aluminium billet to extrusion temperature in 35 minutes, whereas with a conventional hot-air heater the heating time is some fourteen hours. Heating of copper and copper-base alloys presents special problems arising from wide variations in the electrical resistivity and also variations in the thermal conductivity of the material. A typical brass heater rated at 1000 kW will heat some 10,000 lb of metal per hour.

HOW WIDE IS A RANGE?



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● *AUTOMATIC INSPECTION OF MICA INSULATING TAPE*

An interesting recent contribution to industrial inspection technique is a device which detects voids in mica insulating tape and also gives a continuous indication of void area to total area; it can also dye-mark all voids which exceed a pre-selected size. This inspection is carried out before the top layer of paper is applied to the tape, so that corrective measures may be taken before the last manufacturing stage of the tape. In this new device voids in the tape are detected by measuring the current of arcs formed when voids are present in the tape when it is passed between pointed high-voltage electrodes. These electrodes are spaced 0·15 in. above the tape and plane electrodes are in contact with the underside of the tape. A direct-current potential of 7500 volts is applied to the points, so that a current will flow when a void passes between a point and a plane electrode, while no current flows when good mica is between the electrodes. By providing a high-ohm resistor in series with each point electrode, the latter will function independently of all the others. When one point breaks down, the voltage drop due to its current flow is across its associated resistor and high voltage is still applied to the other detector electrodes. An electronic quality indicator circuit is provided, producing a meter reading which is proportional to the current passed and to the ratio of void area to tape area. An alarm circuit is also provided, and this can be adjusted to sound an alarm for any desired reading of the tape quality indicator. It is used to give a warning, should the quality of the tape happen to fall below specifications.

● *COLOURED ENAMEL COATINGS FOR ALUMINIUM*

Vitreous enamel coatings can now be produced on thin aluminium sheet, and these have been found to possess excellent resistance to impact, flexing, and thermal shock. Furthermore, these coatings impart considerable rigidity to the sheet itself. It is claimed that by enamelling an aluminium sheet of about 0·005 in. in thickness, the rigidity of the sheet is increased by fifty per cent. For the successful application of the enamel coating, a clean metal surface is absolutely essential. In the case of lightly oxidised wrought material, degreasing by means of a commercial solvent may be sufficient. Certain heat-treatable alloys require pre-treatment in an alkaline chromate bath. Heavily oxidised material should be submerged for fifteen minutes in an acid-detergent bath, followed by rinsing. Stock thus treated should then be pre-fired, using the same times and temperatures as those employed for the subsequent treatment of the ground coat, and should then be allowed to cool. Normally, two coats of enamel are applied, i.e., an unpigmented ground coat and a coloured cover coat. Application of the enamels may be made by dip, slush, or spray, using standard equipment. After application of the ground coat to the pre-fired material, the coat is allowed to dry and the sheet is fired at 970° to 1000° F for a minimum of five minutes. After cooling to room temperature, the cover coat is applied by spraying and is fired in the same manner as the ground coat. In the case of multi-colour cover coats, each colour is sprayed and fired separately. Enamel thicknesses should range from one to six thousandths of an inch, depending on service requirements. Heavier coatings naturally have a greater resistance to mechanical damage, while thinner coatings have better resistance to thermal shock.

● *PROCESS FOR MAKING TITANIUM CASTINGS*

It is reported that a new process makes the casting of titanium and titanium alloys in intricate shapes a commercial proposition. Hitherto, the greatest deterrent to titanium casting was the lack of a mould material which would withstand the attack of molten titanium. In the new process the pattern is made in the customary way, allowing for shrinkage factors of both metal and mould. A slurry mixture of powdered refractory and liquid binder is poured over the pattern and allowed to harden, to form one half of the mould. The other half is prepared in the same manner. A low-temperature baking of the mould is followed by high-temperature firing, thus producing a strong, cohesive mould. During these steps the mould is given a special treatment, which renders the mould impervious to the action of the molten titanium. After the mould has been equipped with the requisite number of gates and risers, it is placed into the vacuum furnace, and titanium sponge or scrap is loaded into the crucible. The furnace is then sealed and evacuated to one to ten microns Hg, and argon is admitted to provide suitable pressure for melting. An arc is then struck and the melting in the crucible initiated. An observation window enables the operator to note when a suitable quantity of metal has been melted, after which power is cut off and the electrode is moved out of the way. Melt is poured into the gate of the mould, which has been placed directly in line with the pouring lip of the crucible. The mould is cooled in the furnace under argon or vacuum for thirty minutes to one hour, giving the titanium in the mould and crucible sufficient time to solidify and to cool below the temperature at which it reacts with air contaminants.

For further information on developments reviewed in THE BLUE PAGES, readers are invited to write to the Editor.

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